MODERN MECHANICAL ENGINEERING

A PRACTICAL TREATISE
WRITTEN BY SPECIALISTS

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VOLUME III

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HYDRAULICS

BY

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Hydraulics

Physical Properties of Water.—The maximum density of water occurs at 39° F., when, if pure, its weight is 62·42 lb. per cubic foot. The density of sea water varies somewhat with the locality, but its weight may be taken, with a close degree of accuracy, as 64·0 lb. per cubic foot.

Water is slightly compressible, the compressibility varying with the temperature. The modulus of compressibility varies from 368,000 to 288,000 lb. per square inch as the temperature varies from 212° F. to 39° F.

At ordinary temperatures and pressures water is capable of dissolving comparatively large volumes of air. If the temperature is raised, or the pressure reduced, this is liberated, and the hissing, which is so often noticeable where water is escaping at high velocities past the restricted area of a valve seat, is due to the reduction of pressure and the consequent liberation of air bubbles which occurs under these circumstances.

In common with all other fluids, water possesses the property of viscosity. This property, only noticeable when the fluid is in motion, is the cause of all so-called fluid friction, and gives the fluid the appearance of being able to withstand a shear stress between adjacent layers.

Hydrostatics.—In water at rest, the pressure is everywhere the same at the same depth, and is the same in all directions. If W is the weight per cubic foot, the pressure, at a depth h feet, is Wh lb. per square foot, so that a head of water of 1 ft. produces a pressure of 62·4 lb. per square foot, or of ·433 lb. per square inch.

Resultant Pressure and Centre of Pressure.—If p is the mean pressure intensity on a small element of area δa, the force acting on this element will be $p\delta a$, and the total pressure on the whole submerged area will be the sum of all these small normal pressures and will be represented by $\Sigma(p\delta a)$. The resultant of all these elementary forces is termed the resultant pressure on the area, while the point at which the line of action of this resultant cuts the area is termed the centre of pressure.

In the case of a plane surface, the total pressure is the same as the resultant pressure, and is given by $WA\bar{x}$ lb., where A is the area in square feet, and $\bar{x}$ is the depth of its centroid below the water surface, in feet.

The distance from the surface, of the centre of pressure of a plane area is equal to $k^2 \div x'$, where k is the radius of gyration of the area about an

axis in the water surface, and in the plane of the area, and where x' is the distance of its centroid from the water surface, x' being measured in the plane of the area. Thus in the case of a rectangle having one edge in the surface and having its other edges of length l, the value of k^2 is $\frac{l^2}{3}$, and of x' is $\frac{l}{2}$, so that the distance of the centre of pressure from the surface is $\frac{2}{3}l$. If the area is vertical, this is also the depth of the centre of pressure.

Motion of Fluids.—Flowing water may have two types of motion, viz. stream line, and turbulent motion. In stream-line motion the water filaments move in definite and parallel paths, and the resistance to flow is due purely to the shear of adjacent layers and is directly proportional to the viscosity and to the velocity. In turbulent motion the water moves in an eddying mass, the resistance is only to a slight degree dependent on the viscosity and is proportional to the velocity raised to the power n, where n is approximately equal to 2.

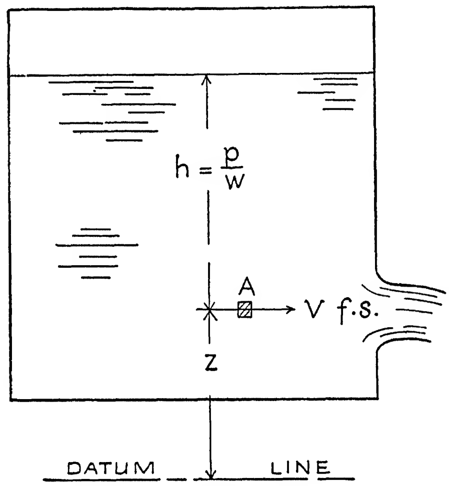


Fig. 1

At very low velocities, the motion is stream line, but as the velocity is increased the motion breaks down and becomes turbulent. For any particular case there is some more or less definite velocity at which the change over from one type of motion to the other takes place, and this is known as the *Critical Velocity*. In pipe flow the critical velocity is inversely proportional to the pipe diameter. At normal temperatures the critical velocity for a 1-in. pipe is approximately ·25 ft. per second, and for the size of pipes usual in engineering practice the motion is always turbulent.

Generally speaking a convergence of the boundaries of a water passage conduces to stream-line motion, and a divergence to eddy formation. Thus in the convergent nozzles of an impulse turbine stream-line motion, characterized by a clear glassy appearance of the jet, is possible at very high velocities.

Water in motion possesses energy in virtue of its velocity, its pressure, and its height. It has kinetic energy, pressure energy, and potential energy. Thus water in motion with velocity v f.s. at the point A (fig. 1) has kinetic energy $v^2 \div 2g$ ft.-lb. per pound. Its pressure energy is $p \div w$ ft.-lb. per pound where p is its pressure in pounds per square foot, and w its weight per cubic foot, and its potential energy is z ft.-lb. per pound where z is its height in feet above some datum level. Each of these expressions is equivalent to a height or head in feet. Thus $v^2 \div 2g$ is the height through which a body falling freely would attain a velocity v, while $p \div w$ is the height of a column

of water which would produce the pressure p at its base. $p \div w$ is therefore called the pressure head.

Bernoulli's Theorem.—If water flows from a point (1) to a point (2), and if there is no loss of energy between these points, the relationship

$$\frac{p_1}{w} + \frac{v_1^2}{2g} + z_1 = \frac{p_2}{w} + \frac{v_2^2}{2g} + z_2 = \text{constant}$$

holds. This is known as Bernoulli's Theorem. If, due to wall friction or eddy formation, there is a loss of energy of h_f ft.-lb. per pound, which is equivalent to a loss of head of h_f feet between (1) and (2), the equation becomes:

$$\frac{p_1}{w} + \frac{v_1^2}{2g} + z_1 = \frac{p_2}{w} + \frac{v_2^2}{2g} + z_2 + h_f.$$

Discharge from a Small Orifice.—Let the suffix (1) refer to a particle of water in the free surface of the reservoir and let (2) refer to the state of affairs in the jet after issuing from the orifice.

Then $\qquad p_1 = 0; \quad v_1 = 0; \quad p_2 = 0.$

$\therefore$ if there is no loss of energy between the surface and the orifice, Bernoulli's equation becomes:

$$z_1 = \frac{v_2^2}{2g} + z_2.$$

or $\qquad v_2 = \sqrt{2gh}$, ft. per second, where $z_1 - z_2 = h$.

Orifice Coefficients.—Actually there is some small loss of energy in the flow towards an orifice, with the result that the true velocity is equal to $C_v\sqrt{2gh}$, where C_v, which is termed the *Coefficient of Velocity*, is about ·98.

Fig. 2

In the case of discharge from a sharp-edged orifice with unobstructed flow towards the orifice, the water filaments overshoot the edge of the orifice, and the section of the jet is less than that of the orifice as shown in fig. 2. The jet becomes parallel at a distance about $\frac{d}{3}$ from the orifice, and its section there is known as the vena contracta. The ratio of this area to that of the jet is termed the *Coefficient of Contraction*, C_c. Its value varies slightly with the size of the orifice and the head, but its mean value, for a sharp-edged circular orifice, is approximately ·62.

The product of the coefficients of contraction and velocity is termed the coefficient of discharge C. Then

$$Q = Ca\sqrt{2gh},$$

where a is the area of the orifice in square feet.

" Q is the discharge in cubic feet per second.

Venturi Meter.—The Venturi meter depends for its principles on the truth of Bernoulli's Theorem. It consists simply of a pipe passing the whole quantity of water to be measured and fitted with a portion BC converging to a throat CD. At D the pipe again diverges to its full diameter. If

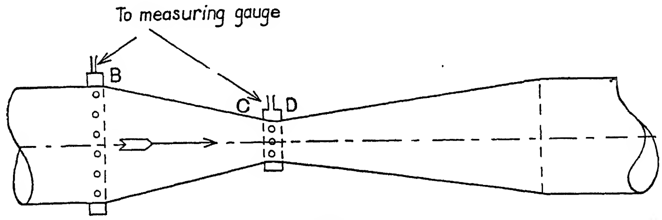


Fig. 3

A and a refer to the entrance and throat respectively (fig. 3), we have, if the pipe be horizontal,

$$\frac{p_A}{w} + \frac{v_A^2}{2g} = \frac{p_a}{w} + \frac{v_a^2}{2g},$$

$$\text{also} \quad v_a = \frac{V_A A}{a}.$$

$$\therefore \frac{p_A - p_a}{w} = \frac{V_A^2}{2g}\left\{\left(\frac{A}{a}\right)^2 - 1\right\}$$

$$\text{or} \quad Q = V_A A = CA\sqrt{\frac{2gh}{\left(\frac{A}{a}\right)^2 - 1}} \text{ c.f.s.,}$$

where h is the difference of pressure at the entrance and throat, measured in feet of water.

In hydraulic problems, atmospheric pressure is taken as the datum pressure, so that, for example, the water in a parallel jet discharging under atmospheric pressure is taken as having no pressure energy.

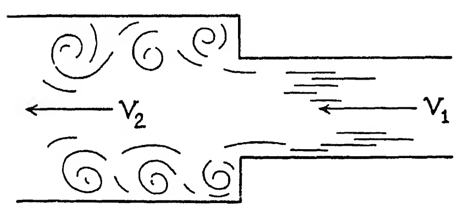


Fig. 4

Hydraulic Losses.—Water flowing over any solid surface, under practical conditions of operation, suffers a loss of energy due to so-called surface friction, which is really due to eddy formation at the surface, and which increases with the roughness of the surface. Also any change in the direction of flow, or any reduction in velocity, such as occurs when the cross-sectional area of a water passage is increased, is productive of eddy formation and of loss of energy.

Loss due to Enlargement of Section.—When a pipe line has its cross-sectional area suddenly increased from A_1 to A_2 sq. ft. (fig. 4), so that the

velocity is reduced from v_1 to v_2 f.s., the loss of head is given very closely by the expression

$$\frac{(v_1 - v_2)^2}{2g} \text{ ft.}$$

This loss may be reduced, within limits, by reducing the velocity gradually (fig. 5). In this case the loss of head is $k(v_1 - v_2)^2 \div 2g$, where k has the following mean values *:

	$\theta°$	2°	5°	10°	15°	20°	30°	40°	50°	60°	70°	80°	90°	120°	150°	180°
Circular Pipe ..	k	·20	·13	·18	·27	·43	·75	·91	1·05	1·12	1·12	1·10	1·07	1·05	1·03	1·00
Rectangular Pipe with one pair of sides parallel ..	k	—	·31	·18	·29	·48	·90	1·10	—	—	—	—	—	—	—	—

These losses include the skin friction in the pipe. This accounts for the value of k increasing as θ is diminished below a definite value, about

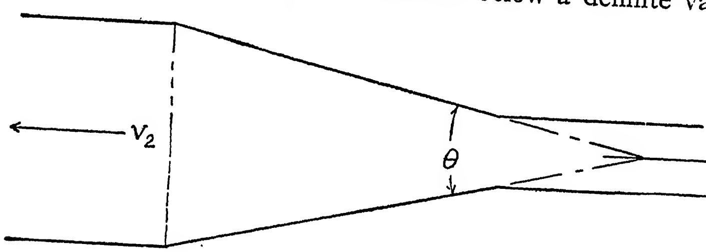
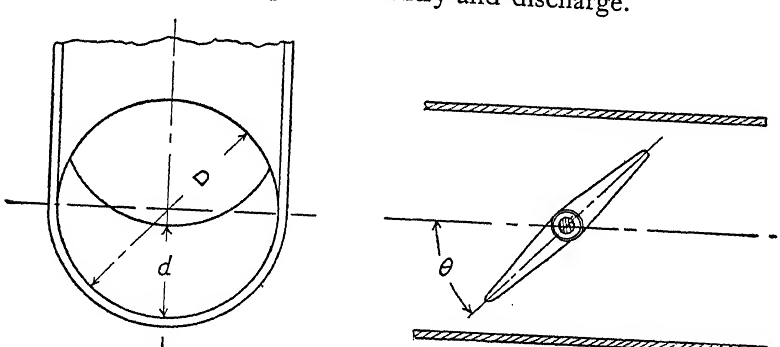


Fig. 5

6° in a circular pipe, and 11° in a rectangular passage, owing to the increasing length of pipe between the points of entry and discharge.



a b

Fig. 6

Loss at Valves and Sluices.—The loss of head due to a partially opened valve is largely due to the expansion of the stream section on passing the constriction. The loss is also affected appreciably by irregularities in

* Gibson, *Hydraulics* (Constable & Co., London, 1912), p. 85.

design, so that values deduced from tests on any one design of valve can only be taken as applying approximately to another valve. The following values have been determined experimentally from valves * of the types shown in fig. 6, *a* and *b*. Here the loss equals $Fv^2 \div 2g$ ft., where v is the velocity in the pipe.

Type of Valve.	$d \div D$									
	·2	·3	·4	·5	·6	·7	·8	·9	1·0	
Circular Sluice Gate (fig. 6*a*), 2″ diam. ..	30	11	4·2	2·1	·9	·35	·22	·07	·00	Values of F.
Circular Sluice Gate (fig. 6*a*), 24″ diam. ..	36	11	3·0	1·6	1·0	—	—	—	—	
	θ									
	5°	10°	20°	30°	40°	50°	60°	70°		
Butterfly Valve (fig. 6*b*)	·24	·52	1·54	3·9	10·8	32·6	118	751	Value of F.	

Losses at Bends in a Pipe Line.—The loss due to a right-angled bend depends on the radius of curvature R of the bend.† The best radius in practice is from 2·5 to 5·0 times the pipe diameter. For such bends the loss is given sufficiently nearly by $\cdot 3v^2 \div 2g$ ft. Where the bend is carried round an angle θ less than 90°, the loss is very nearly proportional to θ^2.

Flow in Pipe Lines.—In designing a pipe line, the problem which usually presents itself to the engineer is that of determining the minimum size of pipe which, with a given loss of head, will discharge a given volume of water per second. The available head is absorbed in giving the kinetic energy of flow in the pipe ($v^2 \div 2g$), and overcoming the pipe line losses which are due:

1. To eddy formation at the entrances to the pipe;
2. To bends, valves, changes of sections, &c.;
3. To wall friction in the pipe.

The loss due to eddy formation at the entrance is small. With a bell mouthpiece it is about $\cdot 05v^2 \div 2g$ ft. With a pipe opening flush with the side of the reservoir it is about $\cdot 47v^2 \div 2g$ ft., and, with a pipe projecting into the reservoir, about $v^2 \div 2g$ ft.

Friction Losses in Pipe Lines.—Many experiments have been carried out to determine the loss due to wall friction in a straight pipe. The earlier experimenters assumed this to be proportional to v^2, and inversely proportional to the hydraulic mean depth m, which is equal to the cross-sectional area $\div$ wetted perimeter.

* Gibson, *Hydraulics* (Constable & Co., London, 1912), p. 249.
† *Ibid*, p. 251.

On this assumption, the loss in friction is written as

$$h = \frac{flv^2}{2gm} \text{ ft.,}$$

or, in the form adopted by Chezy,

$$v = c\sqrt{mi} = c\sqrt{m\frac{h}{l}},$$

where c and f are coefficients whose values depend on the roughness of the pipe. More recent investigations have shown that the coefficient also depends on the pipe diameter and on the velocity of flow, and tend to show that an exponential formula,

$$h = \frac{flv^n}{d^x} \text{ . ft.,}$$

more nearly agrees with experimental results. Values of f, n, and x have been determined by many observers.* Mr. A. A. Barnes,† in a recent discussion of experiments by himself and other observers, gives the following values as applying to new and cleaned pipes:

Material.	Formulæ for		
	Mean vel., f.s. $v =$	Friction head, h ft. $h =$	Q, c.f.s. Q $=$
New uncoated cast-iron pipes	$136{\cdot}6m^{\cdot60}i^{\cdot51}$	$\cdot000343\frac{lv^{1\cdot953}}{d^{1\cdot172}}$	$46{\cdot}7d^{2\cdot60}\times i^{\cdot512}$
New asphalted cast-iron pipes	$174{\cdot}1m^{\cdot769}i^{\cdot529}$	$\cdot000436\frac{lv^{1\cdot891}}{d^{1\cdot454}}$	$47{\cdot}1d^{2\cdot769}\times i^{\cdot529}$
New asphalted single-riveted wrought-iron and steel pipes	$171{\cdot}4m^{\cdot723}i^{\cdot527}$	$\cdot000386\frac{lv^{1\cdot898}}{d^{1\cdot372}}$	$49{\cdot}4d^{2\cdot723}\times i^{\cdot527}$
Do., double-riveted with taper or cylinder joints	$129{\cdot}9m^{\cdot44}i^{\cdot52}$	$\cdot000279\frac{lv^{1\cdot923}}{d^{\cdot846}}$	$55{\cdot}4d^{2\cdot44}\times i^{\cdot52}$
New smooth wood-stave pipes	$223{\cdot}3m^{\cdot66}i^{\cdot586}$	$\cdot00047\frac{lv^{1\cdot707}}{d^{1\cdot126}}$	$71{\cdot}3d^{2\cdot66}\times i^{\cdot586}$
New unplaned wood-stave pipes	$182{\cdot}5m^{\cdot666}i^{\cdot569}$	$\cdot000541\frac{lv^{1\cdot757}}{d^{1\cdot171}}$	$58{\cdot}5d^{2\cdot666}\times i^{\cdot569}$
Clean neat cement pipes	$136{\cdot}3m^{\cdot635}i^{\cdot484}$	$\cdot00024\frac{lv^{2\cdot066}}{d^{1\cdot312}}$	$42{\cdot}0d^{2\cdot635}\times i^{\cdot484}$

Owing to the very convenient form of Chezy's equation,

$$v = c\sqrt{mi},$$

* Gibson, *Hydraulics* (Constable & Co., London, 1912), p. 201.
† A. A. Barnes, *Hydraulic Flow Reviewed* (Spon, London), 1916.

it is often an advantage to have at hand values of c corresponding to various diameters and velocities of flow. Such approximate values are given in the following tables:

Material.	Velocity, f.s.	Diameter Inches.								
		6	12	18	24	36	48	60	72	120
New cast-iron pipes	2	100	107	111	115	120	124	—	—	—
	4	104	111	115	119	124	128	—	—	—
	6	106	113	117	121	126	130	—	—	—
	8	107	114	118	122	127	131	—	—	—
Clean asphalted pipes; smoothly finished concrete pipes and cement-lined tunnels	2	—	103	108	113	120	126	131	135	139
	4	—	108	113	118	126	132	137	141	145
	6	—	112	117	122	131	137	142	145	149
	8	—	115	120	125	134	141	146	149	153
	10	—	117	122	127	136	143	148	151	155
New single-riveted steel or wrought-iron pipes ..	2	—	97	103	108	114	119	123	126	—
	4	—	103	109	114	120	125	129	132	—
	6	—	107	113	118	125	129	134	137	—
	8	—	109	115	121	128	133	138	141	—
	10	—	111	117	123	129	135	140	143	—

Double-riveted steel pipes with cylinder joints have values of c about 5 per cent lower than single-riveted pipes. A new wood-stave pipe has values about 5 per cent lower than a clean asphalted pipe.

After a period of use the incrustation of a pipe line diminishes its discharge. The rate and type of incrustation depends on the class of water and on the material of the pipe walls.

To allow for this diminution, the pipe should be designed to give an initial discharge in excess of the requirements. The excess percentage discharge for different types of pipe should be approximately as follows:

Type of pipe.	Uncoated Cast Iron.	Asphalted Cast Iron.	Asphalted Riveted Wrought-iron or Steel Pipes.	Wood Stave.	Cement or Neat Concrete.
Discharge for which designed, in terms of desired discharge Q	1·55 Q	1·45 Q	1·33 Q	1·08 Q	1·06 Q

Hydraulic Gradient.—If, as in fig. 7, a horizontal AB be drawn through the free water surface, and if ordinates be drawn downwards from AB to represent, on the vertical scale of the drawing, the total loss of pressure

head from the pipe entrance to the particular point considered, the ends of such ordinates, being joined, give a curve called the *hydraulic gradient* for the pipe. If a series of open stand pipes were erected on the pipe line, the free surfaces in these pipes would lie on the gradient line, and the pressure in the pipe is represented, at each point, by its distance below this line. If the pipe is above the gradient line at any point, the pressure will be less than atmospheric. In order to prevent difficulties arising from liberation and

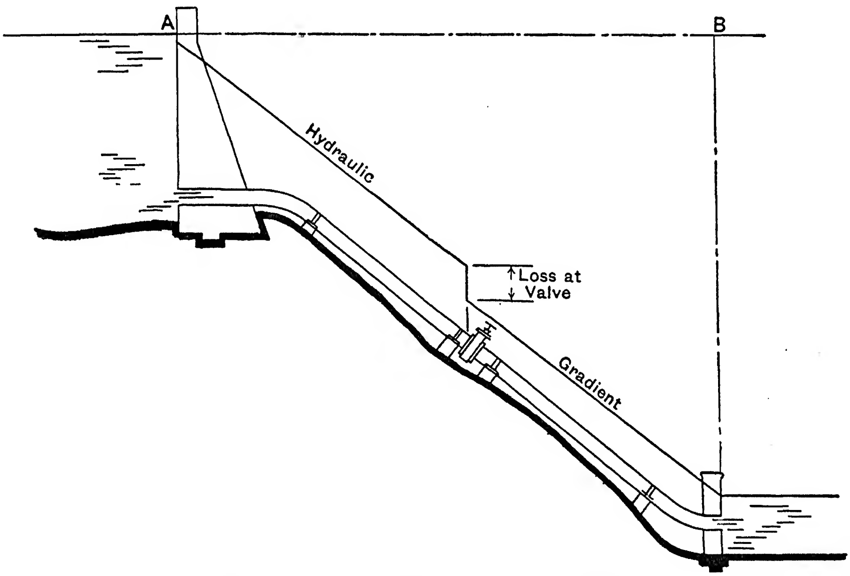


Fig. 7.—Hydraulic Gradient Line of Supply-pipe Line

accumulation of air at such points, and from admission of air at leaky joints, the greatest height above the gradient line should not in any case exceed 20 ft.

Flow through Pipes coupled up in Parallel.—If a series of pipes of diameters d_1, d_2, &c., discharge in parallel between the same two points, so that the available head h is the same in each case, adopting the relationship

$$h = \frac{flv^n}{d^x},$$

the total flow Q, which equals $\frac{\pi}{4}\{v_1 d_1^2 + v_2 d_2^2 + \text{\&c.}\}$ c.f.s., becomes

$$Q = \frac{\pi}{4}\left(\frac{h}{f}\right)^{\frac{1}{n}}\left\{\frac{d_1^{2+\frac{x}{n}}}{l_1^{\frac{1}{n}}} + \frac{d_2^{2+\frac{x}{n}}}{l_2^{\frac{1}{n}}} + \text{\&c.}\right\}$$

$$= \frac{\pi}{4}\left(\frac{h}{f}\right)^{\frac{1}{n}}\Sigma\left(\frac{d^{2+\frac{x}{n}}}{l^{\frac{1}{n}}}\right) \text{ c.f.s.}$$

E.g. taking, as for a cast-iron pipe, $n = 1{\cdot}953$, $x = 1{\cdot}172$,

$$Q = \frac{\pi}{4}\left(\frac{h}{f}\right)^{\cdot 512} \Sigma\left(\frac{d^{2\cdot 60}}{l^{\cdot 512}}\right).$$

Thus three small pipes of diameter d, will give the same discharge as a single large pipe D, of the same length, if

$$3d^{2\cdot 60} = D^{2\cdot 60},$$

$$\text{i.e. if} \quad D = d \times 3^{\frac{1}{2\cdot 6}}$$
$$= 1{\cdot}53d,$$

or one pipe 36·7 in. in diameter would give the same discharge as three 24-in. pipes.

Long Pipe Line, terminating in a Nozzle.—Let A be the area, D the diameter, and V the velocity of flow in the pipe line, and let a, d, and v refer to the nozzle. Thus if h be the available head, and if the Chezy formula be adopted, we have, in a long pipe line:

$$h = \frac{4V^2 l}{c^2 D} + \frac{v^2}{2g}$$
$$= \frac{v^2}{2g}\left\{\frac{8gld^4}{c^2 D^5} + 1\right\}, \text{ since VA} = va.$$

$$\therefore\ v = \sqrt{\frac{2gh}{1 + \frac{8gl}{c^2}\cdot\frac{d^4}{D^5}}} \text{ ft. per second.}$$

In general, the coefficient of velocity, C_v, of a well-designed Pelton wheel nozzle is about ·985, and the velocity will be reduced in this ratio:

Since the energy discharged at the nozzle per second

$$= \frac{wav^3}{2g} \text{ ft.-lb.,}$$

the horse-power delivered at the nozzle is

$$\frac{waC_v{}^3}{2g \times 550}\left\{\frac{2gh}{1 + \frac{8gl}{c^2}\cdot\frac{d^4}{D^5}}\right\}^{\frac{3}{2}}.$$

Accelerated and Retarded Flow in Pipe Lines: Water Hammer.—Where, owing to the gradual stoppage of flow at the lower end of a pipe line, the velocity of the water column is gradually reduced, the retardation being a ft. per second per second, this is accompanied by a rise in pressure at the valve of magnitude $\frac{wla}{g}$ lb. per square foot, or of $\frac{la}{g}$ ft. of water.

If the valve closure is sudden, the elasticity of the water is involved.

Each layer in turn is brought to rest, its kinetic energy is converted into strain energy, and the disturbance is propagated back to the open end of the pipe with the velocity of sound waves through the medium. Under these conditions, the phenomenon is known as *water hammer*, and the rise in pressure p at the valve is obtained from the relationship

$$\frac{v^2}{2g} = \frac{p^2}{2\mathrm{K}w},$$

$$\text{or } p = v\sqrt{\frac{\mathrm{K}w}{g}} \text{ lb. per square foot.}$$

Here K is the modulus of compressibility of the water, which has a mean value of $43{\cdot}2 \times 10^6$ lb. per square foot. Adopting this value, $p = 63{\cdot}7v$ lb. per square inch, a value which shows that excessively high pressures may be obtained with comparatively low velocities of flow, where this action is set up. In a non-rigid pipe line energy is expended in stretching the pipe walls, and the hammer pressure is reduced. Taking this into account, K′, the effective value of K, is given by

$$\frac{1}{\mathrm{K}'} = \frac{1}{\mathrm{K}} + \frac{r}{2t\mathrm{E}}\left(5 - \frac{4}{\sigma}\right),$$

where r is the radius and t the thickness of the pipe, and for steel pipes $\mathrm{E} = 43{\cdot}2 \times 10^8$ lb. per square foot and $\sigma = 3{\cdot}6$.

It may be shown that pressures as great as those corresponding to instantaneous closure are attained if the time of valve closure does not exceed $2l \div \mathrm{V}_p$ sec., where V_p, the velocity of propagation of sound waves along the pipe line, is given by $\mathrm{V}_p = \sqrt{\frac{\mathrm{K}g}{w}}$, and is approximately 4700 ft. per second for a rigid pipe line, but may be as low as 3000 ft. per second for a large thin-walled pipe line. If the time of closure is greater than $4l \div \mathrm{V}_p$ sec., the formula $p' = \frac{wla}{g}$ lb. per square foot is applicable.

Flow in Open Channels.—As in the case of pipe flow, the earlier experimenters assumed the loss of head during steady flow in an open channel to be proportional to the square of the velocity, and adopted one or other modification of the Chezy formula

$$v = c\sqrt{mi},$$

where m is the hydraulic mean depth

$$= \text{cross-sectional area (A)} \div \text{wetted perimeter (P)},$$

and i is the gradient of the channel.

The best-known of these formulæ are due to Ganguillet and Kutter, and to Bazin.

Ganguillet and Kutter put

$$c = \frac{41 \cdot 66 + \frac{1 \cdot 811}{n} + \frac{\cdot 00281}{i}}{1 + \left(41 \cdot 66 + \frac{\cdot 00281}{i}\right)\frac{n}{\sqrt{m}}}$$

and Bazin put

$$c = \frac{157 \cdot 6}{1 + \frac{\gamma}{\sqrt{m}}}.$$

The values of γ and n in these formulæ depend on the roughness of the surface. For straight channels the following values are applicable:

	Character of Surface.	Bazin's γ.	Kutter's n.
A	Smooth cement or planed timber	·109	·009–·010
B	Unplaned timber, slighty tuberculated iron, ashlar, and well-laid brickwork	·290	·012–·013
C	Rubble masonry and brickwork in an inferior condition; fine well-rammed gravel	·833	·017
D	Rubble in inferior condition; canals with earthen beds in perfect condition	—	·020
E	Canals with earthen beds in good condition	1·54	·0225

Bazin's formula appears on the whole to be the more reliable for artificial channels and conduits, and the corresponding values of c in Chezy's formula for the different surfaces and hydraulic mean depths are given in the following table:

Surface.	Values of c computed from Bazin's Formula.				
	$m = \cdot 5$.	$m = 1 \cdot 0$.	$m = 2 \cdot 0$.	$m = 5 \cdot 0$.	$m = 10$.
A	137	142	146	150	152
B	112	122	131	140	145
C	72	86	100	115	125
D	61	74	88	104	115
E	50	62	76	93	106

Form of Channel.—Since in a channel of given sectional area A, the hydraulic mean depth A ÷ P varies with the form of the section, while the resistance to flow increases as A ÷ P diminishes, it becomes important

to determine what form of channel will give the maximum value of A ÷ P for a given value of A, since this will give the maximum discharge for a given slope. Further, as the sectional area of this channel is a minimum, the cost of construction is a minimum, and since in general the perimeter P increases with the area, the cost of lining the channel is also a minimum.

Theoretically the best form of channel is the semicircular section, and for steel and wooden flumes this section is often adopted. For earthen channels the trapezoidal section (fig. 8) with sides sloping at S horizontal to 1 vertical is common, and, for rock channels, the rectangular section. It may be shown that the most economical proportions for such sections are obtained when a circle, with its centre in the water surface, touches the sides and bottom. In a rectangular canal this means that the depth should be one-half the width. In a trapezoidal channel the condition to be satisfied is

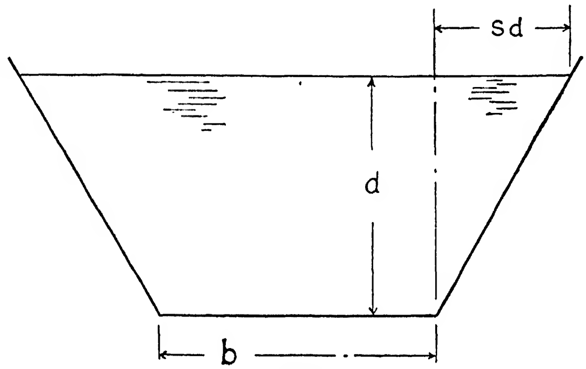


Fig. 8

$$(1 + S^2)d^2 = \left(\frac{b}{2} + Sd\right)^2,$$

where b is the bottom breadth, and d the depth.

The following table shows the top and bottom widths for such a section in terms of the depth, and the depth, perimeter, and hydraulic mean radius in terms of the cross-sectional area.

Side Slopes.	Width. Top.	Width. Bottom.	Depth.	Wetted Perimeter.	Hydraulic Radius.
0 to 1	2·00d	2·00d	·707$\sqrt{A}$	2·828$\sqrt{A}$	·353$\sqrt{A}$
·25 ,, 1	2·06d	1·56d	·744$\sqrt{A}$	2·690$\sqrt{A}$	·372$\sqrt{A}$
·50 ,, 1	2·24d	1·24d	·759$\sqrt{A}$	2·634$\sqrt{A}$	·380$\sqrt{A}$
·75 ,, 1	2·50d	1·00d	·756$\sqrt{A}$	2·640$\sqrt{A}$	·378$\sqrt{A}$
1·0 ,, 1	2·83d	·83d	·740$\sqrt{A}$	2·705$\sqrt{A}$	·369$\sqrt{A}$
1·5 ,, 1	3·61d	·61d	·689$\sqrt{A}$	2·904$\sqrt{A}$	·344$\sqrt{A}$
2·0 ,, 1	4·47d	·47d	·636$\sqrt{A}$	3·144$\sqrt{A}$	·318$\sqrt{A}$
Semicircular	2·00d	—	·798$\sqrt{A}$	2·508$\sqrt{A}$	·399$\sqrt{A}$

Each of these channels has a hydraulic mean radius equal to $d \div 2$. Of the trapezoidal sections, that having side slopes of ·5 to 1 is the most

efficient. The section to be adopted, however, depends also on other considerations. The minimum permissible side slope depends on the character of the soil, and varies from 0 to 1 in rock to 1·5 to 1 in ordinary loamy soil, and 2 to 1 in loose sandy soil. In loose soil a concrete lining enables side slopes of 1 to 1 to be used, and, by preventing erosion of the banks, enables higher velocities of flow to be adopted, while the increased smoothness of the channel enables these velocities to be attained without any greater loss of head. In such a case a concrete-lined channel may be cheaper than one which is unlined.

Velocity of Flow in Open Channels.—The permissible velocity of flow depends on the tendency to erosion of the sides and bed.

Experiment shows that the safe velocity increases with the depth. For medium depths in light soil a mean velocity of from 1·2 to 1·8 f.s. is safe, while in firm loamy soil the safe velocity is from 3·0 to 3·5 f.s. On firm

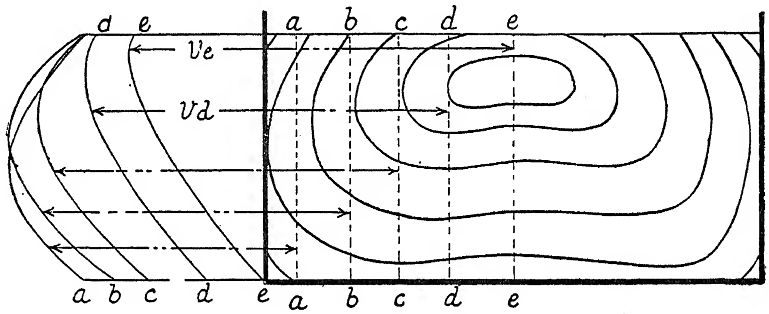


Fig. 9

well-rammed gravel this may be increased to between 5 and 7 f.s. In a concrete-lined channel faced with cement, the maximum safe velocity with water which carries solid material in suspension is about 9 f.s. A higher velocity wears and roughens the bottom until this roughness reduces the velocity sufficiently to prevent further erosion. With a brick or dry-laid heavy rubble channel the velocity should not exceed 15 f.s. Any higher velocity necessitates a carefully-laid facing of heavy masonry with cemented joints.

Distribution of Velocity in an Open Channel.—The distribution of velocity in a straight channel depends somewhat upon the mean velocity. The maximum velocity is found near the centre and in general below the surface, even with a down-stream wind. Its depth usually varies from $\cdot 1h$ to $\cdot 4h$, where h is the depth of the stream. The curves of fig. 9 show typical contours of equal velocity, and the distribution of velocity in a series of verticals in a rectangular channel. Fig. 10 shows the results of a series of gaugings on a concrete channel 16 ft. wide. These curves show the variations of velocity in a vertical plane. The effect of an increase in mean velocity in raising the filament of maximum velocity is well shown by these curves.

It is found that the depth of the point of mean velocity in any vertical is sensibly independent of the direction of the wind. It varies from about $\cdot 55h$ to $\cdot 70h$, depending on the depth and roughness of the channel as indicated below.

Condition of Bed.	Gravel and Small Boulders.				Small Gravel and Sand.				Wood or Cement.			
Depth, feet ..	0 to 2	2 to 4	4 to 6	6 to 10	0 to 2	2 to 4	4 to 6	6 to 10	0 to 2	2 to 4	4 to 6	6 to 10
Depth of point of mean velocity in terms of h.	·54	·58	·62	·66	·57	·60	·65	·69	·61	·65	·68	·70

Generally speaking, the velocity at six-tenths depth in any vertical gives the mean velocity in that vertical within 5 per cent except in abnormal cases,

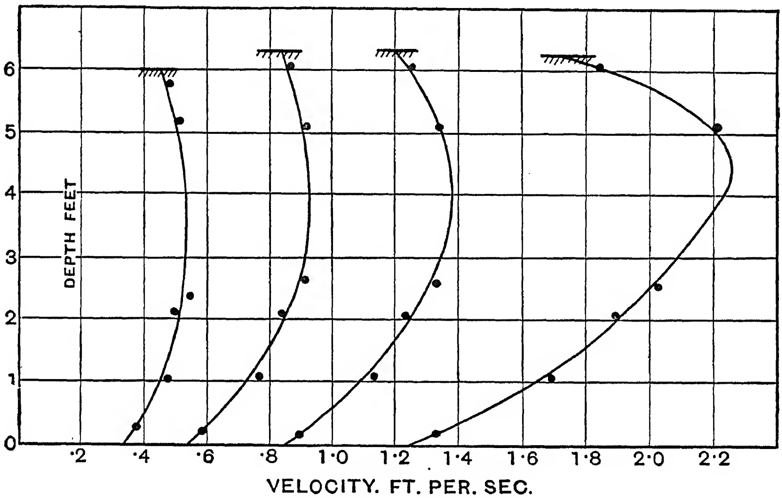


Fig. 10.—Vertical Velocity Curves in Open Channel

while the mean of the velocities at one-fifth and four-fifths of the depth also gives the mean velocity within narrow limits. While the surface velocity should only be used for gauging purposes when other measurements are impracticable, its value, on a still day, is between 80 and 100 per cent of the mean velocity in its own vertical. This factor increases with the depth of the stream and with the smoothness of the channel.

Gauging of Stream Flow.—The method to be adopted in stream gauging depends on the size of the stream, its state, and on the degree of accuracy required.

Where the installation of a weir capable of taking the whole flow is feasible, this forms the most accurate method. For a stream of medium size the rectangular weir is most suitable. For small flows the triangular notch has advantages. For large streams the weir becomes too costly as a temporary measuring device, and if no permanent weir is available the only

way of obtaining the discharge is to measure the mean velocity of the stream and to multiply this by the cross-sectional area. The mean velocity may be obtained in a number of ways.

(*a*) By current meter.
(*b*) By floats.
(*c*) By colour or chemical methods.

Current Meters.—Various types of current meter are in use. Probably the most generally used is the Price meter (fig. 11). The meter is suspended from a rod or cable, and is provided with a guide vane which keeps its axis perpendicular to the direction of the current.

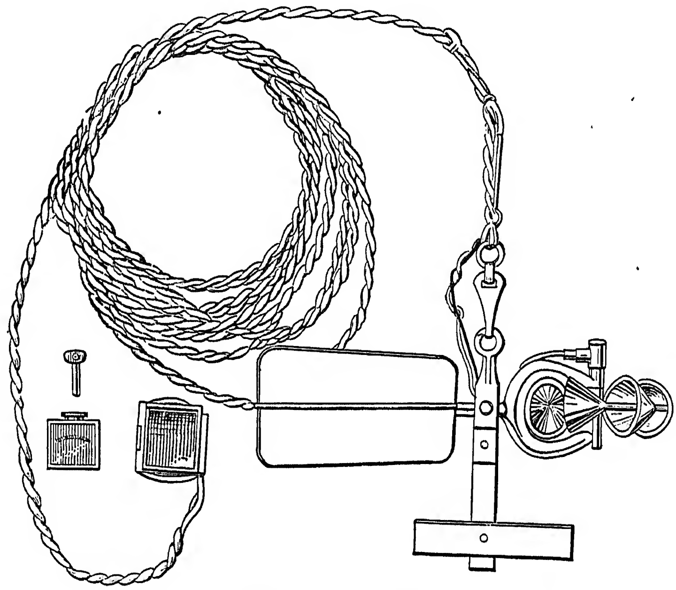


Fig. 11.—Current Meter

The instrument is previously calibrated by towing at known velocities through still water, the number of revolutions corresponding to these velocities being recorded. It has the disadvantages that it cannot be used where floating grass or weed is prevalent, and that it requires rating at frequent intervals. Further, it is unsuitable for very low velocities. The minimum permissible velocity depends on the type of meter, but in general varies from 3 to 6 in. per second.

Meter Observations.—The most usual method of using the meter is called the "point" method, in which it is held successively at certain points in a cross-section. In a shallow stream this may be done by mounting it on a staff which is carried by an observer in waders. In deeper streams it is attached to a heavy sinker, and is suspended from a convenient bridge or from a car carried by a cable across the stream, or from an outrigger fixed to an anchored boat.

In this method, the meter may either be held (1) at several equidistant points in a number of equidistant verticals, the mean velocity being deduced from these readings; (2) at six-tenths, or at mid-depth in a series of equidistant verticals, the mean velocity in each of these verticals then being found by applying a factor; (3) at the surface and bottom only, or at two-tenths and eight-tenths of the depth in a series of verticals; (4) at the surface only. While the first method gives the most accurate results in a steady stream, the length of time necessary to obtain the many observations is a serious drawback, and renders it unsuitable in a stream which is rising or falling.

Generally speaking, the velocity at ·6 of the depth will give the mean velocity in that vertical within 5 per cent, while the velocity at mid-depth multiplied by ·96 will give the mean velocity within about 3 per cent.

Method (3), in which the surface and bottom velocities are measured, is only suitable for shallow streams. Experiments show that the results are fairly accurate if the bed is smooth or gravelly, the depth from ·4 to 1·0 ft., and the velocity from ·5 to 1·5 ft. per second. For deeper streams the mean of readings at $\cdot 2h$ and $\cdot 8h$ is in close agreement with the mean velocity in the vertical, and this method is very often adopted for general stream gauging.

While usually inadvisable to use the surface velocity alone for computing the discharge, it is sometimes impossible in times of flood to make any other measurements. The meter should then be sufficiently submerged to eliminate any disturbance of the surface. Except as affected by the wind, the surface velocity multiplied by a constant which varies from about $\cdot 85h$ in a shallow stream to $\cdot 95h$ in a deep stream gives the mean velocity in a vertical with a fair degree of approximation.

Soundings.—Simultaneously with the meter observations, soundings should be made from which the cross-section of the stream may be obtained.

Float Measurements.—Floats may be divided into three classes: (1) surface floats; (2) sub-surface floats; (3) rod floats.

Surface Floats are liberated at a series of points across the stream at the head of a long straight reach, whose length should be not less than about 200 ft., and the time occupied in covering a measured distance is noted. The surface velocity in each of a number of vertical sections is obtained by repeated observations, and the mean velocity in each vertical is then obtained by multiplying the surface velocity by a factor varying from ·85 to ·95, depending on the depth and condition of the channel. The stream sections may be marked, in a channel of moderate width, by ropes hanging from a bridge or temporary support and trailing in the stream. In a large river this method is impracticable, and observations with the theodolite are necessary to determine the path of the float.

The effect of the wind on the surface velocity renders this method of measurement very unsatisfactory.

Sub-surface Floats consist of bodies having surfaces of large area, as illustrated, for example, in fig. 12, attached to small surface floats for ease of observation, the length of connection being adjusted so as to allow the true float to remain at any given depth. The velocity of the float will then

be approximately that of the current at the required depth. A series of such floats liberated at different points in the cross-section of a stream, the depth of each being ·6 that of the stream at the point of introduction, may be taken as giving the mean velocities in their respective sections. This type is more reliable than the surface float. Experiments show that the errors involved by the use of such floats may be between 5 per cent and 25 per cent.

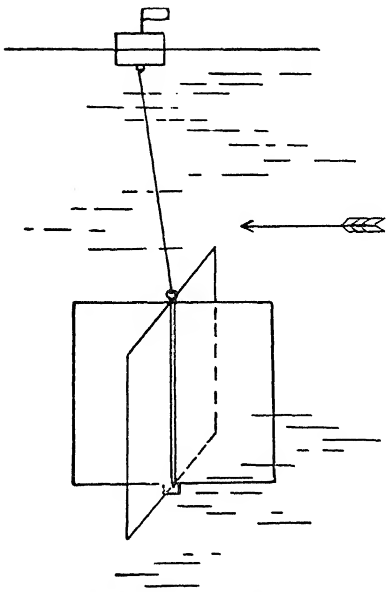


Fig. 12.—Sub-surface Float

The *Rod Float* consists of a light wooden rod or tin tube about 1 in. in diameter, and made in adjustable lengths. The lower end of the bottom section is weighted, and the length adjusted until the rod floats vertically with its lower end clearing the bottom by a few inches. In a large river where these are not likely to interfere with navigation, logs of wood having their lower ends weighted may be used.

The velocity of the rod gives the mean velocity over the vertical in which it floats. The difficulty in using the rod lies in its tendency to drag over shoals and weeds, and to obviate this its lower end may be arranged to float at a height h_1 above the bed of the stream.

For such a case Francis gives the empirical formula

$$v_m = v_r \left(1{\cdot}012 - {\cdot}116\sqrt{\frac{h_1}{h}}\right),$$

giving the mean velocity in the vertical containing the rod in terms of the velocity of the rod v_r, h_1, and h the depth of the stream. Here h_1 should be less than $\cdot 25h$.

In channels of moderate and uniform depth, the rod float is capable of giving results in close agreement with weir gaugings.

Measurement of Velocity by Colour Injection.—The velocity may be determined by injecting colouring matter into the stream, and noting the time this takes to traverse a measured distance. For successful results the colour must be injected in a single burst. In clear water a solution of permanganate of potash may be used. In waters discoloured by organic matter or vegetable stains, red or green aniline dye gives good results.

Gauging by Chemical Methods.—By adding a strong solution of some chemical, for which sensitive reagents are available, at a uniform and known rate into a stream, and by collecting and analysing a sample taken from the stream at some point below, where admixture is complete, the volume of flow can readily be computed.

The method is best adapted to rapid and irregular streams in which the admixture is most thorough, and which, incidentally, are most difficult to gauge by other means.

Flow over Weirs.—The flow over a weir can be expressed as

$$Q = KbH^{\frac{3}{2}} \text{ c. ft. per second}$$

where b is the length of the weir in feet;
where H is the head over the crest, measured to the level of still water behind the weir;
where K is an experimental coefficient, which varies with the type and conditions of discharge.

Sharp-edged Rectangular Weirs.—In the case of a rectangular weir having a thin sharp-edged crest and a vertical up-stream face, the two most useful formulæ are those of Francis and of Bazin.

In the Francis formula $K = 3{\cdot}33$, while b is replaced by $b - 0{\cdot}1nH$, where n is the number of full end contractions. A weir with no end contractions is said to be " suppressed ". In the Bazin formula,

$$K = \left(3{\cdot}25 + \frac{{\cdot}0789}{H}\right)$$

for a suppressed weir. These values of K apply where the area of the approach channel is so relatively large that the effect of the velocity of approach may be neglected. If, as is usually the case in a suppressed weir, the velocity of approach is appreciable, the formulæ become:

$$\text{Francis,}\quad Q = 3{\cdot}33(b - 0{\cdot}1nH)\,\{(H + h)^{\frac{3}{2}} - h^{\frac{3}{2}}\} \text{ c.f.s.}$$

$$\text{Bazin,}\quad Q = \left\{1 + {\cdot}55\left(\frac{H}{P + H}\right)^2\right\}\left\{3{\cdot}25 + \frac{{\cdot}0789}{H}\right\}bH^{\frac{3}{2}} \text{ c.f.s.,}$$

where, in the Francis formula, $h = v^2 \div 2g$, and where v is the mean velocity in the approach channel, while, in the Bazin formula, P is the height of the weir crest above the bed of the channel.

The above formulæ apply only to a weir having free access of air to the under side of the falling sheet or nappe. If the nappe clings to the crest or front face of the weir, or if free access of air is prevented, the discharge is increased.

Triangular Weirs.—If the weir is thin-crested and sharp-edged, and if θ be the angle between its two sides,

$$Q = 4{\cdot}28\, c \tan\frac{\theta}{2} \,.\, H^{\frac{5}{2}} \text{ c.f.s.,}$$

where c depends slightly on θ, and H is measured in feet.

$$\text{If } \theta = 90^\circ,\ c = {\cdot}593, \text{ and } Q = 2{\cdot}536\, H^{\frac{5}{2}} \text{ c.f.s.}$$

$$\text{If } \tan\frac{\theta}{2} = 2,\ c = {\cdot}618, \text{ and } Q = 5{\cdot}29\, H^{\frac{5}{2}} \text{ c.f.s.}$$

Cippoletti Weir.—If the sides of a weir having two end contractions be inclined outwards at an angle θ with the vertical (fig. 13), the value of K in the formula $Q = KbH^{\frac{3}{2}}$ is sensibly independent of the head if θ is such that the side slope is 1 horizontal to 4 vertical. Such a weir is called a *Cippoletti weir*. The discharge is given by

$$Q = 3{\cdot}37\, bH^{\frac{3}{2}} \text{ c.f.s.}$$

if the velocity of flow in the approach channel is negligible, and by

$$Q = 3{\cdot}37\, b\{(H + h)^{\frac{3}{2}} - (h)^{\frac{3}{2}}\} \text{ c.f.s.},$$

as in the Francis formula, when the velocity of approach is taken into account.

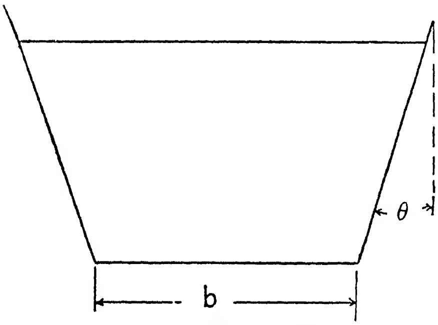


Fig. 13.—Cippoletti Weir

Broad-crested Weirs.—Experiments indicate that if the width of the crest of a sharp-edged weir is less than about ·33H, the nappe will spring clear of the crest. Weirs with wider crests, in which the nappe adheres to the crest, are termed broad-crested weirs. Expressing the discharge over such a weir as

$$Q = K'bH^{\frac{3}{2}},$$

values of K′ have been determined experimentally for a very large number of weir sections.*

Precautions to be adopted in Weir Gaugings.—The standard sharp-edged weir having a free discharge, or, for small quantities, the right-angled triangular notch, are the only types for which the coefficients have been determined with sufficient accuracy to admit of use for accurate measurement of flow without previous calibration.

For accurate measurement the following are essentials:

1. Sharp-edged weir sill, fixed so as to be incapable of vibration, having its face vertical and perpendicular to the direction of the stream, and, if rectangular, having its sill horizontal.
2. Clear discharge into air, with no adherence of the vein to the weir face.
3. Weir long in proportion to its depth, i.e. $b > 3H$.
4. H small in comparison with the depth of the approach channel, and sectional area of vein (bH) not greater than one-sixth that of this channel.
5. Suitable channel of approach. This should be as long and of as uniform section as possible so as to allow of the motion becoming steady before reaching the weir. The length should, if possible, exceed 30H, this ratio being increased where the length of weir is largely in excess of 3H.
6. Accurate determination of the head H. For accurate work the surface-level should not be taken in the stream itself, but in a stilling box or pit from 18 in. to 2 ft. square communicating with the stream through a pipe

* *Hydraulics* (Gibson), Constable & Co., London, 1912, p. 163.

of about ɪ in. diameter. The zero of the gauge should be accurately adjusted to the level of the weir crest. For accurate work, where individual readings are to be taken, a hook gauge (fig. 14), provided with a vernier for reading to the nearest ·001 ft., and with screw adjustment, is best.

Impact of Jets.—In the case of the impact of a jet on a stationary or moving surface, the force exerted in any direction is equal to the rate of change of momentum per second in that direction.

Impact on a Fixed Surface.—

Let a = sectional area of jet in square feet,
,, v = initial velocity of jet in feet per second.

Then the weight of water impinging on the surface per second = wav lb.

The initial momentum of this in the direction of motion

$$= \frac{wav^2}{g} \text{ ft.-lb. per second.}$$

If θ be the angle through which the jet is diverted

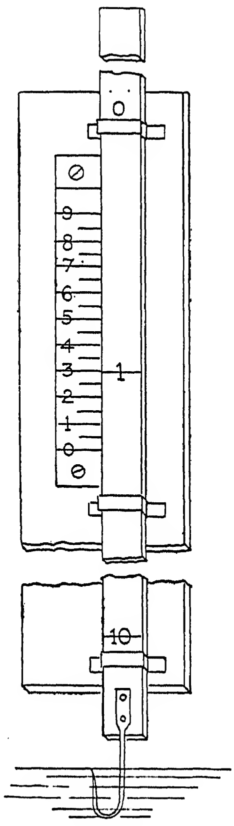


Fig. 14.—Hook Gauge

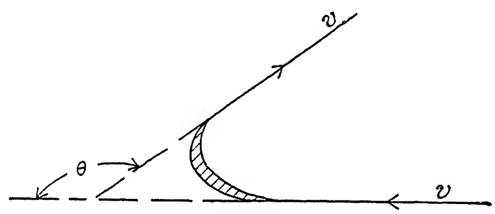


Fig. 15

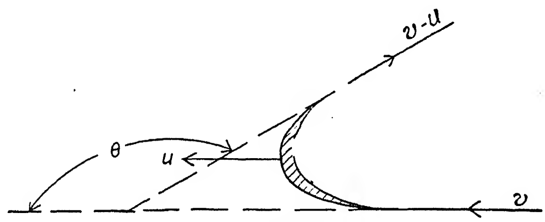


Fig. 16

(fig. 15), and if the relative velocity of the water and the vane is unaffected by the impact, the final velocity in the original direction will be v cosθ, and the final momentum will be $\frac{wav^2}{g}$ cosθ.

∴ Change of momentum in this direction per second

$$= \frac{wav^2}{g}(1 - \cos\theta)$$

$$= \text{Force exerted in pounds.}$$

Impact on a Series of Moving Vanes.—(Fig. 16). If the vanes are moving

in the original direction of motion of the jet, with velocity u f.s., the relative velocity of the water and the vane is $v - u$, and the final absolute velocity of the water in the original direction of motion is

$$u + (v - u) \cos\theta.$$

The change of velocity in this direction is then

$$v - u - (v - u) \cos\theta = (v - u) \{1 - \cos\theta\},$$

and the change of momentum per second, or the force on the vanes, is given by

$$\frac{wav}{g} \cdot (v - u)\{1 - \cos\theta\}.$$

The work done on the vanes

$$= \frac{wavu}{g} (v - u)\{1 - \cos\theta\} \text{ ft.-lb. per second.}$$

Differentiating with respect to u, it appears that this expression is a maximum when $u = \frac{v}{2}$, or when the velocity of the vanes is one-half that of the jet. If the effect of frictional resistances is taken into account, the best velocity for the vanes is slightly less than $\frac{v}{2}$, and in the case of a Pelton wheel, the ratio of u to v is usually between 0·45 and 0·47.

If $u = \frac{v}{2}$, the work done on the vanes

$$= \frac{wav^3}{2g} \left\{\frac{1 - \cos\theta}{2}\right\} \text{ ft.-lb. per second,}$$

and since the kinetic energy of the jet per second is equal to $\frac{wav^3}{2g}$ ft.-lb., the efficiency is equal to $\frac{(1 - \cos\theta)}{2}$

When $\theta = 180°$, this equals unity, while if $\theta = 90°$, the efficiency is 0·5.

MODERN PUMPING MACHINERY

BY

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Modern Pumping Machinery

INTRODUCTORY

The term " Pumping Machinery " embraces a surprisingly large variety of appliances ranging from simple hand-operated devices to elaborate power-driven machines of immense size. Water may be raised by wind power, animal effort, the energy of running streams, falling water, compressed air, hydraulic power, jets of water, steam jets, combustion of gas, the pressure of steam, by shock pressure (" hydraulic slam "), centrifugal pressure, direct lift, and, combined with the introduction of suitable mechanical apparatus, by the electric current, and any other form of motive power. Within the limits of this article it is only possible to deal with the principal types of power-operated pumps in common use, and to give particular notice to the three essentially modern forms, viz. (1) the quick-running reciprocating pump, (2) the " high duty " slow-speed reciprocating type, and (3) the high-pressure multi-stage turbine pump.

Classification of Pumps.—Pumping appliances may be classified under three main divisions:

1. *Reciprocating Pumps*, including: lift pumps of various forms; plunger or force pumps, single and double acting of many kinds; bucket and plunger, piston and plunger or plunger and plunger, sometimes called " differential ", pumps of various kinds; diaphragm pumps; semi-rotary pumps; balers.

2. *Rotary Pumps.*
   - (*a*) Pure velocity forms: centrifugal and turbine pumps; axial flow (propeller) pumps; screw or helical pumps.
   - (*b*) Positive displacement forms: cog or gear pumps; sliding blade, eccentric drum pumps; chain pumps; band pumps.

3. *Fluid Impellent Pumps.*
   - (*a*) Pulsating forms: Savery type steam pump; hydraulic rams; pneumatic pumps; gas (explosion) pumps.
   - (*b*) Continuous flow: air lift (emulsion) pumps; jet pumps; injectors.

RECIPROCATING PUMPS

Various Types.—A reciprocating pump in its simplest form consists of a cylindrical barrel provided with inlet and outlet openings, suitably controlled by valves, and a reciprocating member working within. All the important variations are shown diagrammatically in fig. 1, *a* to *i*. The sketches show vertical pumps, but each type can be arranged horizontally by altering the branch connections.

"Lift" or "bucket" pumps *a* and *b* are those in which the water is sucked through a foot valve, or suction valve, on the ascending stroke, and driven through the bucket on the descending stroke.

Piston, plunger, or force pumps, *c*, *d*, *e*, *f*, have a solid reciprocating member which alternately sucks through a suction valve and discharges through a delivery valve.

Differential pumps, *g*, *h*, and *i*, are single-acting on the suction side, but deliver water on both strokes—for this reason they are sometimes termed "two-stroke" pumps.

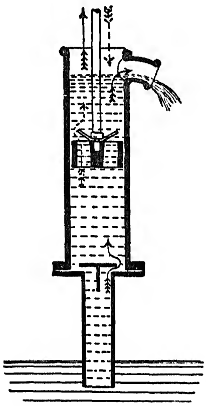
Single-acting pumps give an intermittent discharge, while double-acting and differential pumps give a fluctuating discharge without a definite stoppage of flow. A group of three single-acting pumps delivering in regular rotation into a common main or a pair of double-acting pumps similarly adjusted give a nearly uniform discharge.

While it is inadvisable to generalize too much concerning the application of these various types, it may be said that. as a rule:

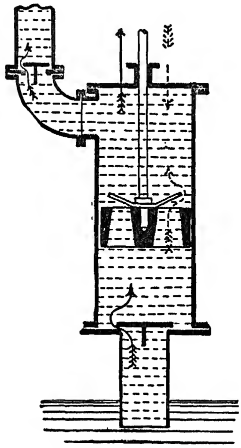
(1) bucket pumps (*a*, *b*, *g*) are generally confined to well, mine-shaft, and bore-hole pumping, and to large drainage pumps;
(2) single-acting piston pumps, *e* (usually made with a trunk piston), are generally used in groups of two or three side by side for pumping against moderate heads, with transmission drive;
(3) single-acting ram or plunger pumps, *c*, are used similarly, but for heavier pressures;
(4) the double-acting piston pump *f* is the recognized type for direct-driven steam pumps for moderate heads;
(5) the double-acting ram pump *d* similarly for direct-driven steam pumps for heavier pressures; and
(6) the differential pumps *h* and *i* are often adopted on quick-running pumps—or for slow-speed pumps delivering into a very long pipe main.

Some important modifications to the double-acting ram pump should be noticed. For high pressures it is usual to make this type either with two opposed rams, fig. 2, *a*, connected by yoke rods on each side of the barrel, thus eliminating one stuffing box and rendering the remaining two very accessible, or, the rams are arranged as in fig. 1, *d*, but operated through a trunnion pin attached to the centre of the ram between the stuffing boxes.

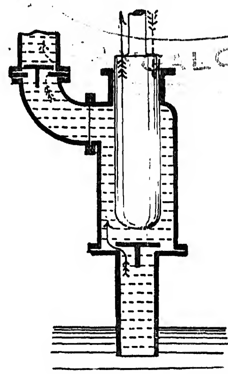
S.A. = Single acting.
D.A. = Double acting



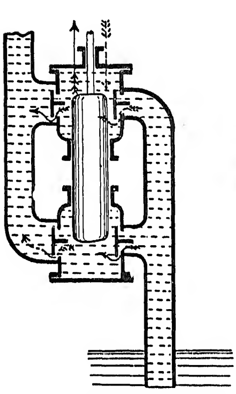
(*a*) Suction or bucket pump, S.A.



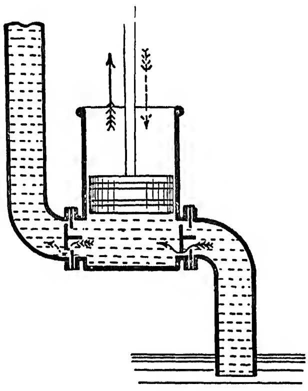
(*b*) Lift, suction and force, or bucket pump, S.A.



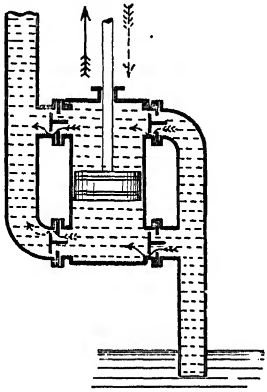
(*c*) Ram, plunger, or force pump, S.A.



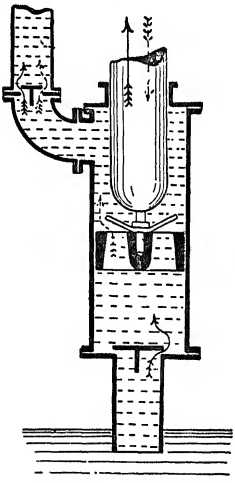
(*d*) Ram or plunger pump, D.A.



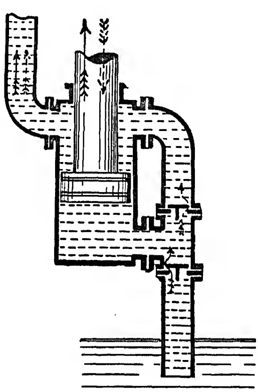
(*e*) Piston or force pump, S.A.



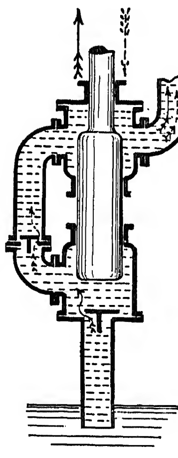
(*f*) Piston pump, D.A.



(*g*) Bucket and plunger, or differential pump, S.A. on suction side, D.A. on delivery side



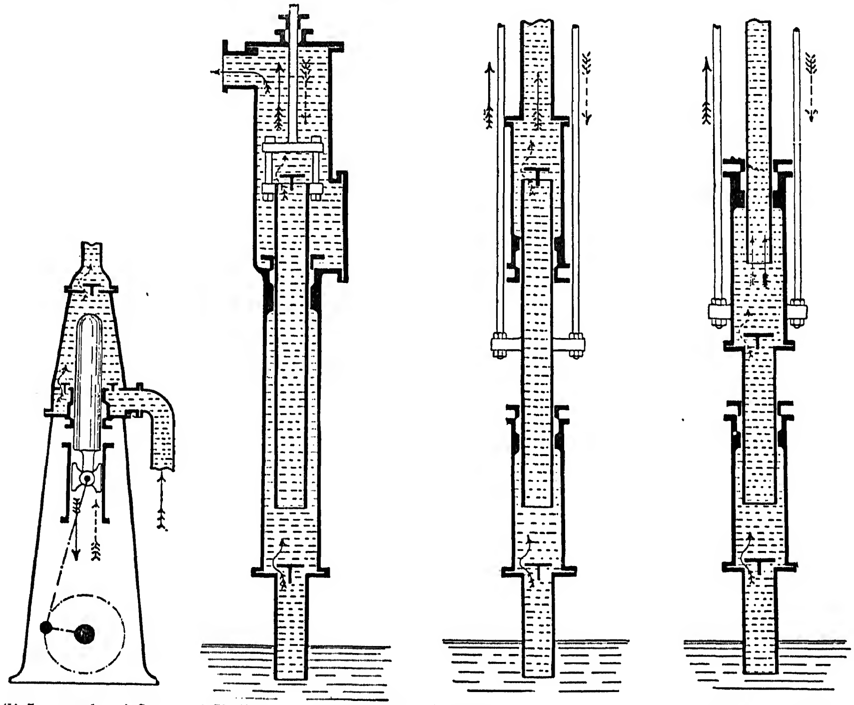
(*h*) Piston and plunger, or differential pump, S.A. on suction side, D.A. on delivery side



(*i*) Plunger and plunger, or differential pump, S.A. on suction side, D.A. on delivery side

Fig. 1.—Types of Reciprocating Pumps

Another modification of the ram pump has resulted in a " uni-flow pump " which is made in various forms both horizontal and vertical, fig. 2, *b*, and also in the differential form. Here the suction valve is usually of annular



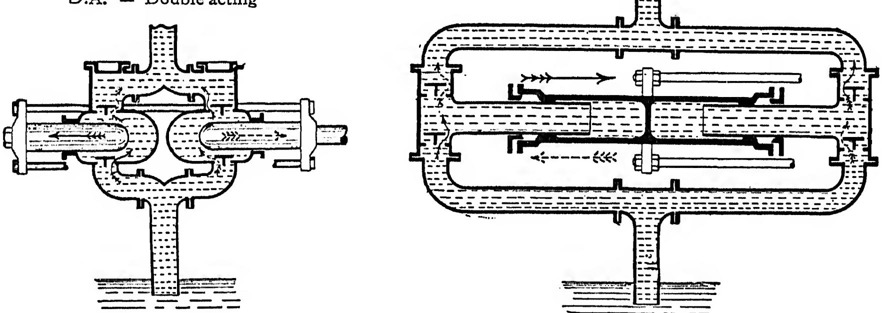
(*b*) Inverted uni-flow pump, S.A.

(*c*) Hollow ram pump with internal rod, S.A.

(*d*) Hollow ram pump with external rods, S.A.

(*e*) Differential hollow ram pump, S.A. on suction side, D.A. on delivery side

S.A. = Single acting
D.A. = Double acting



(*a*) Horizontal D.A. ram pump (opposed plungers)

(*f*) Horizontal D.A. hollow ram pump with oscillating barrel

Fig. 2.—Special Varieties of Pumps

form encircling the ram, and the special feature of the design is that flow takes place in one direction only in the pump chamber, this direction coinciding with the direction of the thrust-stroke of the ram.

A further interesting variety is the hollow ram pump which is made in several forms, fig. 2 (*c*, *d*, *e*, and *f*). Its action is, of course, identical with

that of the bucket pump, a hollow ram sliding through packed glands taking the place of the bucket.

Reciprocating Pump Diagram.—The conditions in the working barrel of a reciprocating pump are best studied with the assistance of a diagram of work done by the pump. Such diagrams (fig. 3) are taken by means of an indicator connected with the pump cylinder, the drum being rotated in the usual way by a string attached to the crosshead. The diagram shows the conditions in a single-acting pump or on one side of the piston of a double-acting pump. The suction stroke commences at A, where the

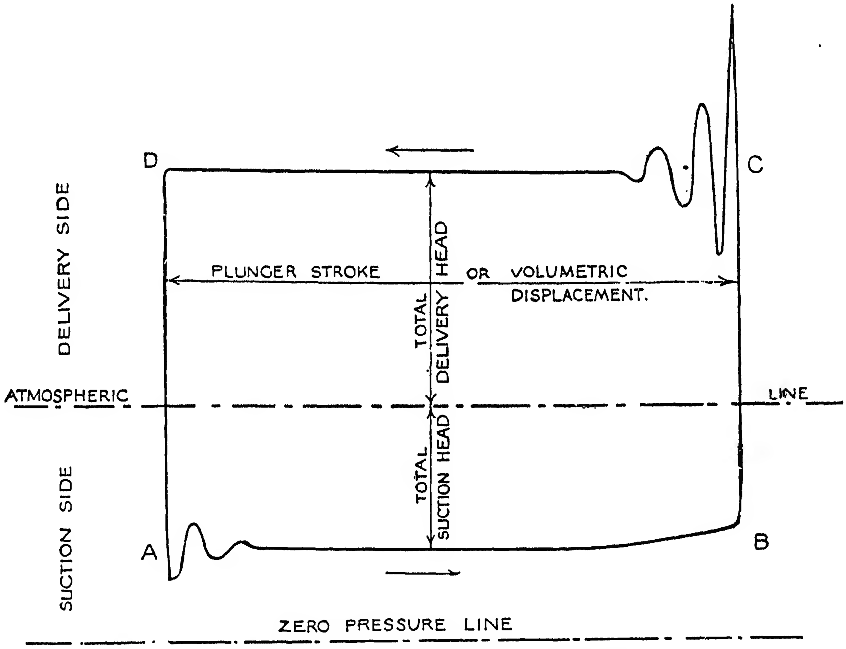


Fig. 3.—Indicator Diagram from Reciprocating Pump

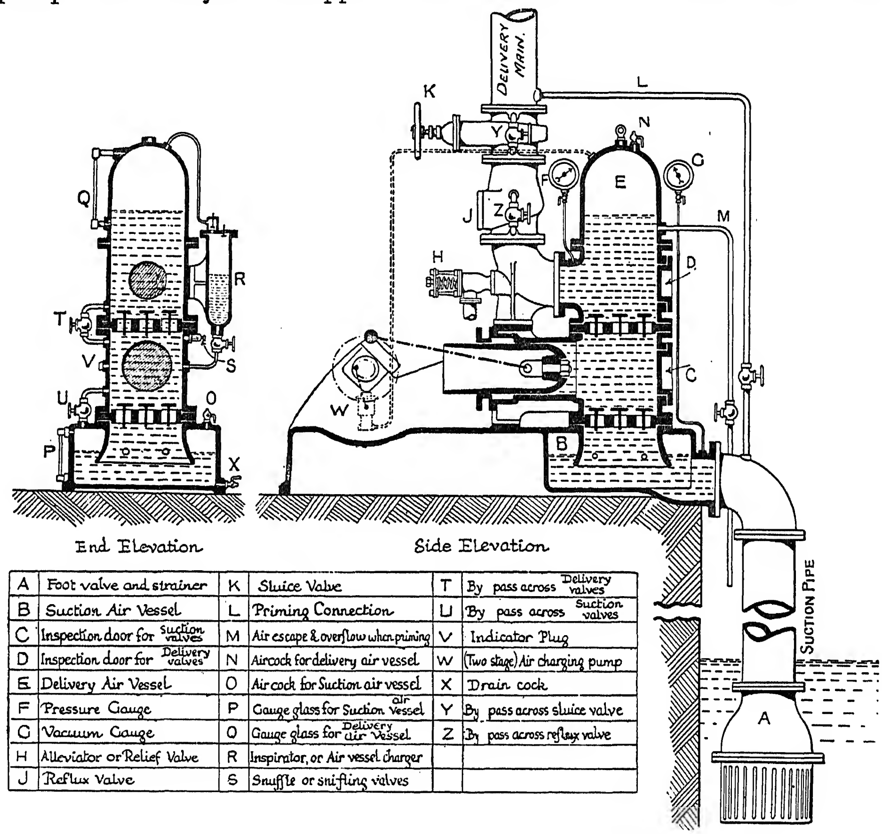
wavy line is produced by the inertia forces due to setting the suction column in motion. During this stroke the plunger draws water by overcoming a negative head equivalent to the suction height to be lifted plus the frictional head incidental to the flow in the suction main, plus the head necessary to maintain the velocity in the system. As the suction stroke approaches completion and the plunger begins to slow up, the retardation of the suction water may cause a rise in the diagram towards B. The plunger then reverses and acts against the total delivery head. The effect of this reversal is shown by the oscillating inertia pressures at C. The pressure stroke is then completed against the full delivery head to the point D on the diagram where the plunger comes to rest, ready for the suction stroke and so to repeat its cycle.

The area of such a diagram is a measure of the actual work done by the pump per stroke.

Typical Arrangement of Pump, Accessories, and Fittings.—It

is unnecessary to describe the fittings and accessories for each type of pump separately, as these details will vary according to the duty and type of the pump. All ordinary fittings and accessories (apart from such mechanical necessities as oil-pumps, lubricators, &c.) are shown on the reference diagram, fig. 4.

Each fitting is, of course, made in many different forms, and no single pump would carry all the appurtenances shown, for the reason that several



A	Foot valve and strainer	K	Sluice Valve	T	By pass across Delivery valves
B	Suction Air Vessel	L	Priming Connection	U	By pass across Suction valves
C	Inspection door for Suction valves	M	Air escape & overflow when priming	V	Indicator Plug
D	Inspection door for Delivery valves	N	Aircock for delivery air vessel	W	(Two stage) Air charging pump
E	Delivery Air Vessel	O	Air cock for Suction air vessel	X	Drain cock
F	Pressure Gauge	P	Gauge glass for Suction air vessel	Y	By pass across sluice valve
G	Vacuum Gauge	Q	Gauge glass for Delivery air vessel	Z	By pass across reflux valve
H	Alleviator or Relief Valve	R	Inspirator, or Air vessel charger		
J	Reflux Valve	S	Snuffle or snifting valves		

Fig. 4.—Reference Diagram for Pump Accessories and Fittings

of them are alternative devices. The diagram is self-explanatory, but calls for a few remarks.

A *foot valve* at the end of the suction pipe is not always a necessity, though when combined with a priming device it always hastens commencement of pumping after starting up. Many modern pumps are designed with a large clearance volume in the pump barrel, and with these pumps, as well as in all cases of high suction lift, a foot valve and priming apparatus is generally advisable.

In most modern equipments a *suction air vessel* will generally be fitted. This accessory has two main functions, (1) to maintain a uniform velocity of flow in the suction pipe, so avoiding water hammer, cavitation, or inertia

shocks, and (2) to reduce the valve shock when the suction valves close.

The volume of an ordinary suction air vessel, as usually supplied with commercial pumps, will generally be found to be about two and a half or three times the displacement of one plunger per stroke. Such an accessory is quite suitable for the majority of pumping schemes where the suction lift does not exceed 12 to 14 ft., and where at the same time the suction pipe is not long or tortuous. For greater suction lifts the volume ratio would be increased.

The *delivery air vessel*, if placed immediately over the delivery valves, has a very important cushioning effect upon all the shocks in the pump, and promotes silent running. If, however, it is separated by branches and passages from the pump chamber it can only have an equalizing effect upon the velocity in the delivery main, thereby reducing pipe shocks, &c. Where several large pumps, or pump barrels, deliver into a common main it is not uncommon to provide a large air vessel for the main and smaller vessels over each set of delivery valves. In practice, air vessels range in volume from 3 times to 30 times and in exceptional cases as much as 100 times the displacement of the pump plunger per stroke.

For very high pressures it is customary to use a powerful spring-loaded (sometimes steam loaded or otherwise) *pressure equalizer* on the discharge pipe instead of an air vessel.

The *alleviator* or *relief valve* is the safety valve of the system, and will save the pump from damage in the event of the sluice valve being closed at starting or if any obstruction gets into the mains.

With reciprocating pumps it rarely happens that both a *reflux valve* and a *sluice valve* are fitted at the same time. Either will serve the purpose of isolating a pump from a common main if dismantling is necessary. When a reflux valve is fitted, an air vessel is advisable on the pipe line at the outlet side of the reflux valve.

The object of the *overflow*, fig. 4, M, is to ensure that the delivery air vessel is not filled with water when priming.

Water under pressure has the property of readily dissolving air, and a delivery air vessel will become water-logged, and therefore useless, in a very few hours if not constantly replenished with a fresh supply of air. There are many ways of recharging an air vessel. On very large pumps an independent air compressor is usually provided, and on smaller pumps either an inspirator of some kind or a small air pump is fitted, driven from some moving part. Sometimes the small air pump is arranged to draw air from the suction air vessel—where there is usually more air than is needed—and discharge into the delivery air vessel.

An *inspirator* is really an air pump so arranged as to utilize the main pump plunger and the full pressure of the pump for compressing air into the delivery air vessel. The type shown in fig. 4 is automatic when brought into use by opening the stopcock communicating with the pump barrel. On the suction stroke air is then drawn in through the small atmospheric valve, and on the delivery stroke it is forced through the small outlet valve

into the air vessel. The water chamber forming the body of the inspirator, though an improvement, is not a necessity; also for small pumps a simplified form of hand-controlled device may be adopted.

A *snuffle* or *snifting valve* is sometimes fitted to admit air under the delivery valves for the purpose of replenishing the delivery air vessel. This is not a desirable accessory, as it reduces the volumetric efficiency of the pump, and it can only be tolerated on quite small pumps on account of its simplicity and convenience. If used it will often silence a noisy pump, but at the expense of loss in pumping efficiency.

By-pass valves are used either for priming, for relieving a pump of pressure when starting up, or for draining the different chambers of the pump.

In designs providing separate valve boxes, or valve pots, for the suction and delivery valves it is customary to fit pet cocks to the valve boxes in order to blow off air.

Reciprocating Pump Practice

Power Pumps.—The term " power pumps " covers the largest class of pumps and includes all those which are not direct-driven by some prime mover, i.e. all pumps operated by transmission drive of some kind, such as by pulley and belting or by gearing. To distinguish between varieties it is customary to speak of (*a*) slow-running or quick-running, (*b*) long-stroke or short-stroke pumps. Such distinctions are only relative, and it is impossible to define rigidly what these terms mean. Generally speaking a slow-running pump would run from 20 to 70 r.p.m., a normal pump from 50 to 100 r.p.m., a quick-running pump from 70 to 120 r.p.m., all according to the length of stroke. A very quick-running or " express " pump would run from 130 to 300 r.p.m., according to the length of stroke. Also, a slow-running pump would usually be a long-stroke pump, a normal or ordinary quick-running pump would be of normal proportions, and an express pump would be a short-stroke pump. By " long stroke " would be meant a pump in which the stroke was from 2 to $2\frac{1}{2}$ or more times greater than the bore; a normally proportioned pump would have a stroke either equal to the bore or only some 25 per cent greater; while a short-stroke pump would have a stroke less than the bore, and down to so small an amount as $\frac{1}{2}$ to $\frac{1}{4}$ of the bore. Generally speaking the longer the stroke of a pump the better it is from the hydraulic point of view, because higher pumping speed is obtained with fewer reversals of direction and consequently fewer shocks and less " slip " at the valves. A short-stroke pump is the outcome of a necessity for higher rotary speeds of the pump crank-shaft. The mean piston speed is not necessarily any higher with a quick-running pump than with a slow-running pump, and in fact it will generally be found that the highest piston speeds occur on large long-stroke pumps. A pump as usually designed for normal or for slow running would be quite unsuitable for running at a high rotary speed, as special provision has to be made for dealing with the quick reversals of direction.

The mean piston speed of a pump is always a factor of special interest.

It is mainly on account of the great inertia of water that, while piston speeds of 500 to 700 ft. per minute are common on steam-engines, and speeds of over 1000 ft. per minute are regular practice in certain forms of engine, yet with pumps the mean piston speed of common practice is but 100 ft. per minute, whilst 250 to 350 ft. per minute represents the limit of general practice, and higher speeds of 400 to 600 ft. per minute only occur as rare and special examples. Power pumps are either fitted with fly-wheels or there is some equivalent weight in the gear wheels or pulley, or there is a uniform driving torque which maintains their rotary speed uniform. Their actual piston speed therefore varies from zero to a maximum and back again to zero every stroke. The relation between the mean piston speed and the maximum speed is influenced by the length of the connecting rod, but for almost any practical purpose it is permissible to assume that the rod is of infinite length, in which case the maximum piston speed is $\frac{\pi}{2}$ times the mean piston speed, or say 57 per cent higher. It is important to remember therefore that, as all the water velocities are controlled by the piston, the maximum velocities—for example, through the valves and water passages —are 57 per cent in excess of the mean velocity as usually calculated from the total pump displacement.

We can now distinguish between the characteristics of the slow-running and quick-running varieties of power pumps.

Slow-running and Normal Power Pumps.—These are made in vertical and horizontal forms, and each barrel is usually single-acting. The vertical form has the advantage that it occupies very little floor space, and the horizontal form has the merit of great rigidity and absence of vibration.

Other considerations are that in the vertical form the whole working stress falls on the shaft bearing cap and bolts, whilst on the horizontal type these stresses are borne by the bedplate itself. As regards durability of the plunger and barrel, the vertical form has the advantage that wear takes place evenly all round, and that when grit is present it is not so liable to reach the glands. Due to the weight of the plungers, the wear on glands, barrel, and plunger on a horizontal pump takes place in a downward direction. Grit, if present, also settles to the bottom side and still further contributes to wear.

Single- and double-barrel types are only used for unimportant duties and the standard form is the treble-barrel, three-throw, or triplex power pump. The reason for this is fairly obvious. We referred in the previous section to the great fluctuation of the piston speed during each stroke when the pump was driven at a uniform rotary speed, and it is clear that a single-barrel pump would produce a series of somewhat violent pulsations or shocks in a delivery main alternating with intervals of rest. A double-barrel pump, with cranks at 180° apart, would produce the same pulsations, but at double the frequency and without the interval of rest. In a treble-barrel pump, however, the cranks are set at 120° apart, and the pulsations overlap one

another and produce a combined discharge which is practically uniform in pressure and quantity.

A further important consideration in electrically-driven pumps is that the motor requires a uniform torque in order to operate successfully, and this result is best given by a three-barrel pump. Hence the general adoption of the triplex form of power pump. Such pumps are usually proportioned with the stroke longer than the bore to the extent of 25 to 100 per cent. The piston speeds vary from 60 ft. per minute on small pumps to 120 ft. per minute on large sizes, with a corresponding number of reciprocations from 100 per minute on quite small sizes to 20 or less per minute on the largest sizes. They are rarely made larger than 12 or 14 in. in the bore, additional sets being installed when larger quantities are required. The mean water velocity through the lifted valve is commonly between 2 and 4 ft. per second, though on large high-pressure pumps velocities as high as 10 or even 12 ft. per second are sometimes allowed in order to reduce the valve lift as much as possible.

Fig. 5 shows a few typical valves as commonly fitted to slow-running power pumps, but there are an immense variety of these details in practice and doing excellent service. The valves shown at *a* and *b* (fig. 5) are the forms generally adopted when " multiple-valves " or " valve groups " are employed. The " lift " is generally about $\frac{1}{2}$ in., and the size of discs from 3 to 4 in. diameter, the valve-port circle being $\frac{1}{2}$ in. in diameter less than the disc. The spring load on these valves is generally light. A common amount for suction valves is to allow a force of 1 lb. per square inch of net port area to lift the valve $\frac{1}{4}$ in. off its seat, and for delivery valves the spring would be of two or three times this stiffness. Valve *c* is a heavier and stronger type often used as a single valve in small pumps, and as a multiple valve on larger pumps. It is rarely made in sizes larger than about 5 in. diameter, as the valve-port area then bears an extravagant relation to the valve-lip area, and a ring with double lip then becomes advantageous. Simple ring valves are shown at *d* and *e*, and a large multiple ring form at *f*. The heavy valves *g*, *h*, and *i* of double beat pattern are used on waterworks, deep well, or mine pumps, and will be referred to later.

Barrels, plungers, and valve boxes are made of cast iron, unless specially corrosive liquids or other requirements demand gun-metal. When necessary, barrels or plungers are generally gun-metal lined or covered in preference to being made entirely of gun-metal; smaller diameter rams are often solid bronze. Easy access to valves is a feature of all sound design.

Entirely on the score of costs, various changes are made in the method of supporting the crank-shaft and the crosshead according to the water pressure for which the pump is designed. Thus, for low pressures, the crank-shaft will only have two supports, one at each end, in the form of substantial plummer blocks; but for high pressure, the crank-shaft will have a support between each barrel.

Again, there will be no crosshead (as usually understood) for quite low pressures, a trunk ram being quite capable of dealing with all side thrust,

whilst in some cases where the diameter of the ram is small, an extension rod is added to the ram, and works in a small guide placed between the crankshaft and crosshead.

For higher pressures a bored trunk guide is provided to carry the crosshead for both horizontal and vertical forms of pump, and for the highest

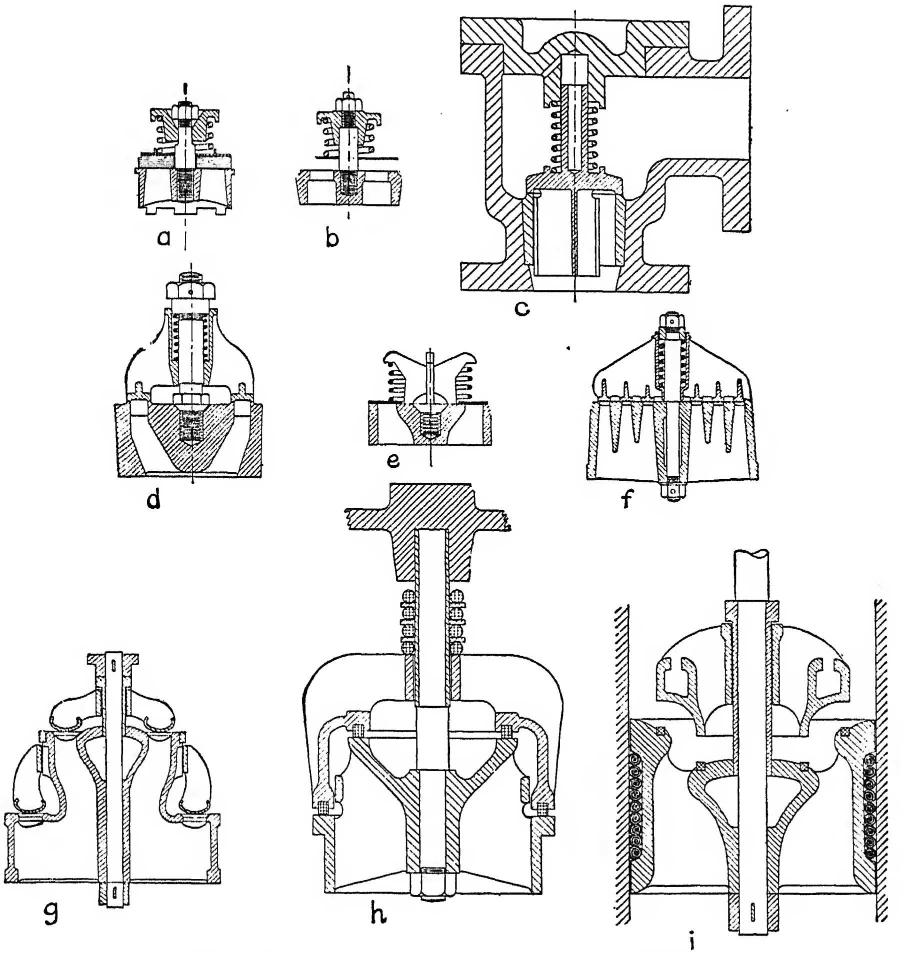


Fig. 5.—Typical Valves for Slow-running and Normal Power Pumps

pressures—usually occurring on horizontal pumps only—flat guides take the place of the trunk guide.

Single reduction gearing will be used for low and moderate pressures, though this feature is controlled largely by the speed of the available source of drive.

A few typical illustrations will show the wide variety of types embraced by the term power pumps. Fig. 6 shows a simple vertical pump built by Messrs. Hayward, Tyler, & Co., Ltd., London and Luton, and is typical

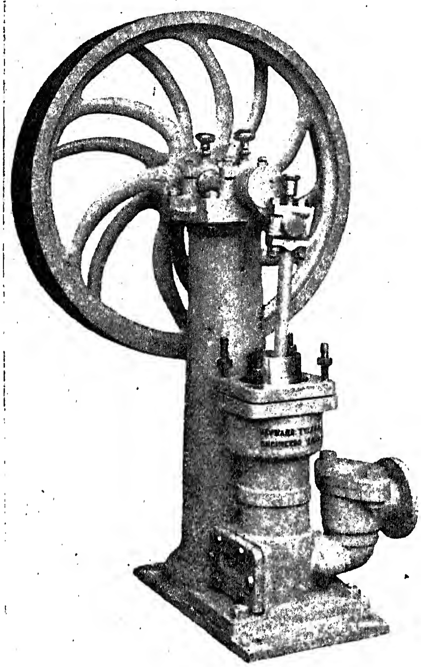


Fig. 6.—Hayward-Tyler Single-barrel Vertical Pump

of cheap small-capacity belt-driven pumps for low and moderate pressures. Such pumps would have cast-iron bodies, plungers, and glands; the suction and delivery valves and seats would be of gun-metal, and the fly-wheel may be turned to take the driving belt.

Fig. 7 shows a double-barrel double-acting geared horizontal pump suitable for pulley drive, and also built by Messrs. Hayward, Tyler, & Co., Ltd. The pump barrels, valve boxes, and covers are of cast iron, the barrel being brass lined, and the pistons are of brass, fitted with white-metal rings. These pumps are usually fitted with flat disc-type valves similar to fig. 5, *b*.

An example of a three-throw trunk-ram pump built by Messrs. Glenfield & Kennedy, Ltd., Kilmarnock, is shown in fig. 8. This shows a vertical low-pressure triplex design with two crank-shaft bearings and a single reduction gearing. The rams are of hard cast iron and the valves of best quality rubber, spring loaded, closing on gun-metal seatings.

Other varieties of power pumps for more severe duties are illustrated and discussed under the headings of hydraulic and deep-mine pumps.

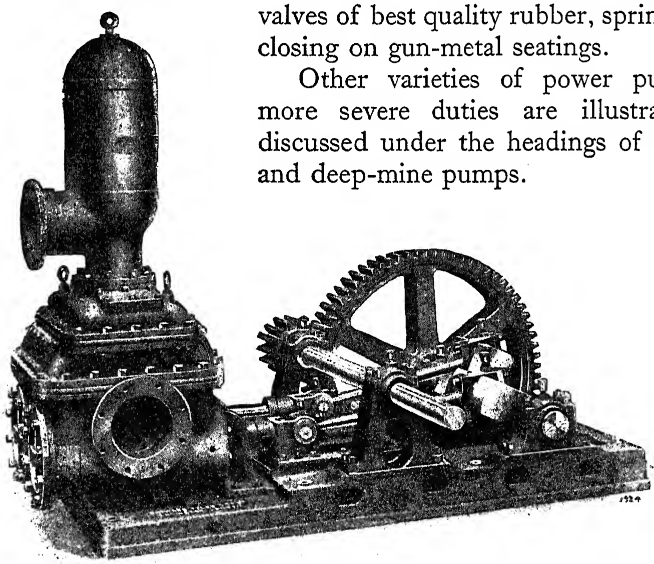


Fig. 7.—Double-barrel Double-acting Horizontal Pump

Fast-running Power Pumps.—Intermediate between a fast-running pump and a slow-speed pump is a type which is merely a slightly modified normal pump speeded up to run at from 30 to 50 per cent faster than the slow-running kind. The considerations necessary to convert a slow-running pump into one for a moderately increased speed are identical with those for designing a quick-running pump, but the principles are carried out to a less degree. It is only necessary, therefore, to consider the features essential for a successful fast-running power pump.

A quick-stroke pump will have:

1. Light yet strong valves, so designed that the valve loading is made up of the minimum possible dead-weight and the maximum possible spring load to produce the required loading, thereby eliminating as much inertia as possible.

2. A very small lift to the valves, so reducing their motion as much as possible to facilitate rapid opening and closing. The mean velocity through the lifted valve (across the beat) is frequently carried higher than with slow-speed pumps, but will not generally exceed 10 ft. per second for the delivery valve and 8 ft. per second for the suction valve.

3. A free and easy flow from the inlet branch to outlet branch, particularly between suction and delivery valves and plunger chamber. As a rule the plunger oscillates in a large water chamber immediately between the suction and delivery valves. This large water volume acts as a cushion to damp inertia forces, and materially assists in preventing "separation" of the water from the ram at high plunger velocities.

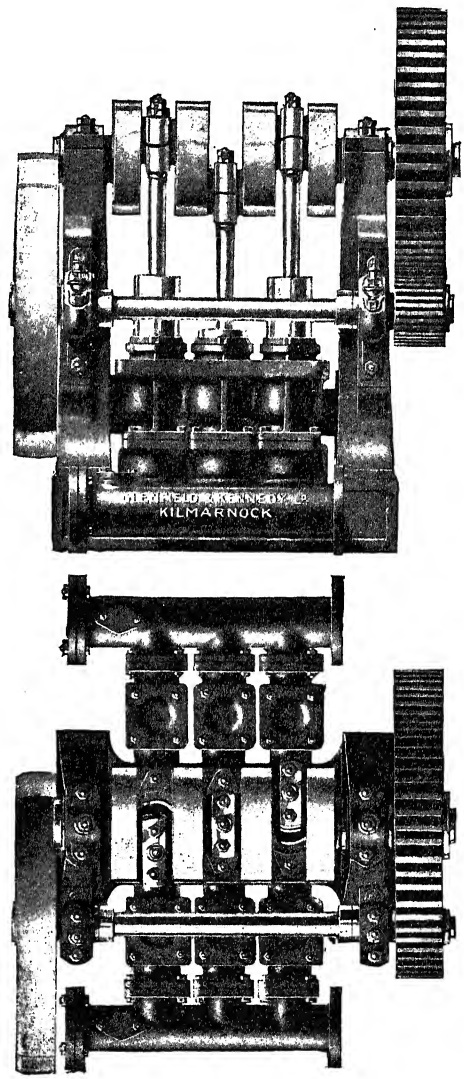


Fig. 8.—Vertical Low-pressure Triplex Pump, Two Bearings, Single-gear Reduction, Trunk Rams

4. Large volume suction and delivery air vessels in close relation to the respective valves.

5. A moderately high mean piston speed. For average size pumps this will vary from 250 to 400 ft. per minute, and on long-stroke pumps of comparatively slow reciprocations speeds as high as 500 ft. per minute have been successfully attempted.

The best material to use for valves and valve seats to withstand the hammer of opening and closing and the erosion of the water is necessarily a matter of careful study. For this purpose special bronzes are generally selected for severe duties or good gun-metals for average work. Much attention has also been given to the material of the valve springs, and india-rubber is in considerable favour as opposed to steel or bronze.

The mechanical design of a quick-stroke pump requires more careful treatment than a slow-speed pump. All the inertia forces of moving parts are, of course, greater and the rubbing velocities are faster at all pins and guides. Naturally the allowable working pressures on crank pin, crosshead pin, main bearings, and crosshead guide will be less on a quick-speed pump. Efficient lubrication becomes of greater importance, and on a high-class pump automatic lubrication by means of an oil pump or mechanical lubricator is a necessity.

The horizontal form of pump is customary for the quick-stroke type on account of its greater rigidity, lower centre of gravity, and larger base for securing to foundations.

There are several general types of high-speed pumps, and also several well-known special designs. Fig. 4 represents diagrammatically the general features of the simplest form of quick-stroke pump. In that form they are made with double, or generally triple, barrels. Similarly, but with opposed yoked rams (see fig. 2, *a*), they are made in double crank units, so composing a four-barrel pump.

As a single-barrel pump the ram is usually made in two diameters, and the design modified (on the lines of fig. 1, *i*) to produce a differential or "plunger and plunger" pump.

Fig. 9 shows several forms of valves as used on quick-running pumps. The commonest type is some form of ring valve, similar to fig. 5, *f* or fig. 9, *a* or *b*, but several special designs of valve have become associated with quick-revolution pumps. The valve shown at *a* (fig. 9) has separately seating rings which can adjust themselves to each annular seat, and should any foreign matter become lodged on any one seat the one ring only is prevented from properly closing. Metal to metal faces are used when clean water is being pumped, but should it be necessary to deal with gritty or dirty water a special seating material is employed. The valve at *b* is designed for dirty water, and here the seat is composed of two materials, a metal ring which takes the pressure load on the valve and a leather or dexine, &c., strip which provides an efficient joint even though grit be present. With such valves it is usual to allow a water velocity of only one-third to one-fourth of the velocity possible with an all-metal valve.

The valve shown at *c* is a modification of the well-known "hat-band" valve· in this case a series of rubber rings carry out the double function of

sealing the seat and providing the necessary spring load for closing. Example *d* is a special metallic valve, originated by Messrs. Hoerbiger, of Vienna, and sometime used by Messrs. A. Borsig, Berlin, and is remarkable for its lightness. The resilience of the material of the disc provides the required spring load, but in cases of large valves this is supplemented by fitting light springs over the valve seat.

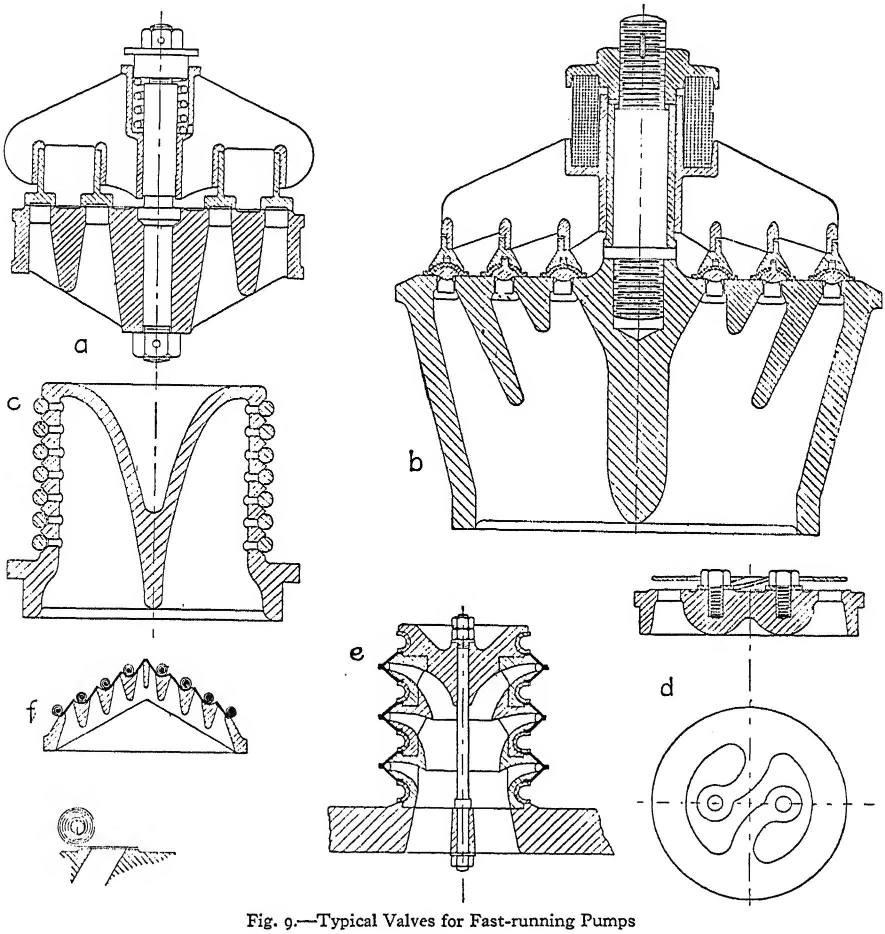


Fig. 9.—Typical Valves for Fast-running Pumps

The annular lip valve *e*, or Witting's valve, is made by Messrs. Balcke, Bochum, and Messrs. Grevenbroich, Germany, and takes several forms according to the requirements of the pump design. The valve lips are dished bronze rings held in place and spring loaded by a rubber retaining ring, this ring also at the same time making an effective joint at all the points of possible leakage.

In the Gutermuth valve *f* is seen an interesting development of the common spring-loaded flap valve. This valve is exceedingly sensitive, and when opened offers an unobstructed flow through the valve port.

Practical examples of horizontal quick-stroke pumps as made by Messrs. T. H. and J. Daniels, Ltd., Stroud, England, and fitted with Gutermuth valves, are shown in figs. 10 and 11.

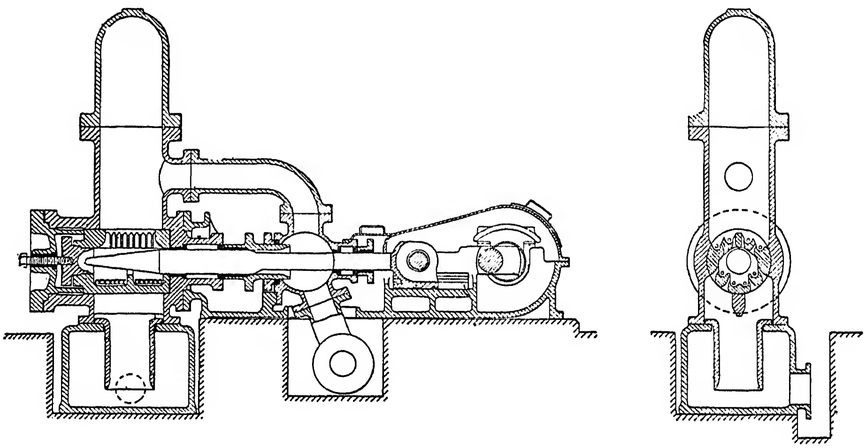


Fig. 10.—Single-barrel Differential Horizontal High-speed Pump

Fig. 10 shows a section of a single plunger differential pump for heads of 800 ft. and over. The moving parts are enclosed and provided with automatic lubrication, and the speeds vary from 150 to 200 r.p.m. according

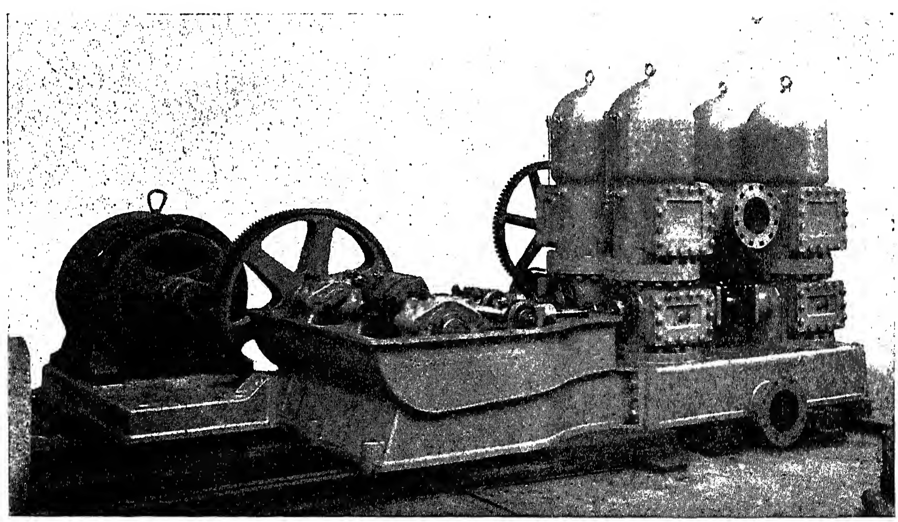


Fig. 11.—Motor-driven, Four-barrel Double-crank Horizontal High-speed Pump

to the pump size. An illustration of a four-barrel double-crank pump is given in fig. 11. The plant shown has plungers $5\frac{13}{16}$ in. diameter by 12 in. stroke, and delivers 30,000 gall. an hour against 500 ft. head, and running at 100 r.p.m.

In general features these illustrations represent the designs of most quick-revolution pumps, though departures are made by each maker to suit the particular type of valve adopted.

Open-well Pumps.—Well pumps may be divided into two classes: those in which the pump is placed on a platform down the well and is therefore accessible by means of a ladder or lowering rope, and those in which the pump is contained in a tube far beyond the reach of an attendant. The former class are known as open-well pumps and the latter as tube-well or bore-hole pumps. In order to obtain an economical output for the size of well, and from the pumping equipment, the natural limitations in horizontal dimensions are counteracted as far as possible by providing long-stroke bucket pumps, similar in design to fig. 1, *b*, and made up in single-, double-, or triple-barrel units. The single-acting cylinder, on account of its cheapness and smaller space occupied, is used for all small and moderate size wells, but in the case of large waterwork pumps double-acting forms are always preferred. The commercial open-well pump is understood to be an equipment with pump barrels, usually of gun-metal, arranged near the water surface, plungers of the bucket type, vertical pump rods leading to the surface, and a head gear consisting of crank-shaft and gears, connecting rods, and guides.

Tube-well or Bore-hole Pumps.—A simple tube-well pump consists of a foot valve, a bucket plunger with spear rods, and a crank-shaft head gear. The bore-hole lining, or tube, often forms the rising main, and a special length of this tube, near the lower extremity, acts as the pump barrel, but in most cases there is a separate tube, forming the pump proper, arranged inside the bore-hole lining. For these pumps, depths as great as 250 ft. are not uncommon, though naturally the great weight of the vertical rods necessary for such depths is a factor requiring special study. Tube-well pumps are usually single-acting and the useful work, both suction lift and delivery pressure, is therefore done on the up-stroke only; during the down stroke the bucket and vertical rods simply fall through the water in the vertical tube. Obviously, therefore, with single-barrel or single-crank pumps, the falling parts require to be balanced so as to reduce the shock in falling, and also to reduce the dead-weight to be lifted on the up-stroke. The necessary balancing is generally effected in two ways, the one hydraulic, the other mechanical. The hydraulic method is to provide a plunger, on the spear rods, working through a gland disposed either at the ground level or at a convenient platform above the highest level to which the water ever rises in the bore-hole. This is often called a " differential plunger " and, in effect, converts the pump into a " bucket and plunger pump ", similar to fig. 1, *g*, but with a very much extended barrel. Fig. 12 shows a complete arrangement diagram of such a pump as made by Messrs. Hayward, Tyler, & Co., Ltd., London. The area of the plunger is generally made equal to half the area of the bucket, consequently on the up-stroke the pump only delivers half the volume of its displacement, and the other half-volume is delivered by the displacement of the plunger on its down-stroke. Thus

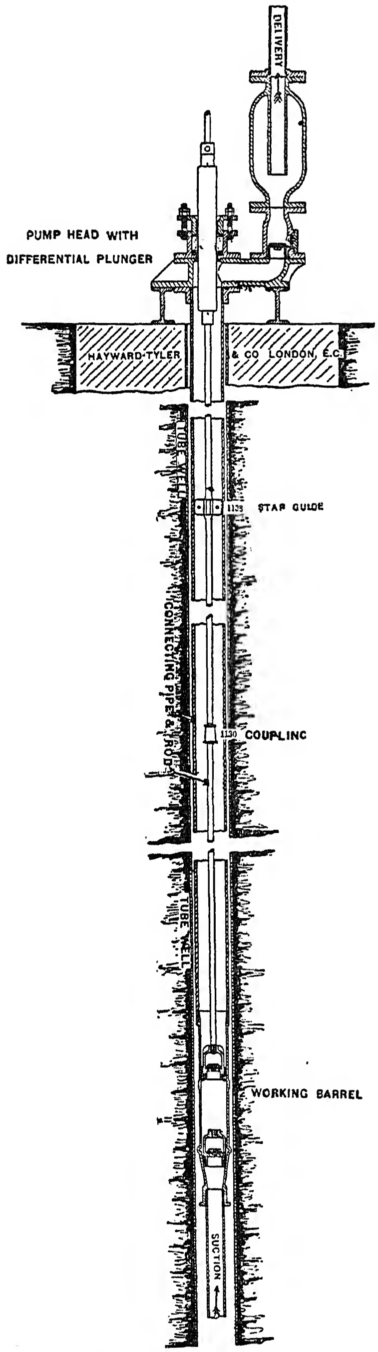


Fig. 12.—Tube-well Pump

the fall of the spear rods is cushioned by doing work on the down-stroke, and the torque on crank-shaft is partially equalized for the same reason. The mechanical portion of the balancing is carried out by connecting a counter-balancing weight to the vertical rods at the surface. Tube-well pumps are made in sizes from $1\frac{1}{2}$ in. to 12 in. bore and with strokes from a few inches to 3 or 4 ft., practically the same type of pump being developed to larger sizes in connection with pumping engines for waterworks purposes.

On account of its intermittent action, a single-acting unit is frequently incapable of extracting the full quantity which a given bore-hole is capable of discharging. To meet this condition a double-acting type of tube-well pump is necessary. The form usually provided is that known as the "concertina type". This consists of an arrangement (fig. 13) with two buckets and without a foot valve. The upper bucket is usually attached to a tubular rod, and the lower one to a rod passing through the tube of the upper one. For correct operation one bucket must be coming up while the other is going down, and a special head gear is necessary, as shown on the illustration, with a pair of parallel cranks to drive the tubular rod B and a single opposite crank for the centre rod A, or alternatively an opposed bell-crank lever system. The behaviour of the pump is easily followed. As the buckets separate, the upper bucket discharges water through the vertical delivery tube while on its under side it sucks water through the lower bucket, which for the time being acts as a foot valve; then, as the buckets approach each other, the lower bucket delivers water through the top bucket, and merely elevates the suction water below it. Thus water is discharged on both strokes, and both suction and delivery water columns are in motion during both strokes. In addition to extracting

more water from the bore-hole, this pump has the advantage that all the valves are on the buckets, which can be easily raised to the surface—the tedious operation of groping for a foot valve is entirely eliminated—and, further, there is no need of balance weights, as the work done is equal on both up- and down-strokes, thus giving a steadier torque.

A limitation in capacity is placed on all the tube-well pumps so far described, owing to the restriction in possible valve area in the buckets and the foot valve, and because the maximum pump speed must be determined by the maximum permissible water velocity through the valves. The Ashley pump was designed by Mr. Herbert Ashley about twenty-two years ago, to provide larger valve areas in relation to the bore-hole area, and thus permit of higher bucket speeds, and a consequently greater delivery of water for a given size of bore-hole. It combines the further advantage that a foot valve is not required either on the single-acting pump or the double-acting form. The licensees for the Ashley pump are Messrs. Glenfield & Kennedy, Ltd., Kilmarnock, and fig. 14 shows an example of their single-acting design. In this form (fig. 14) the suction valves are seen embodied with the bucket, and the barrel is closed in at the bottom; the action is quite obvious from the drawing. In the concertina type of Ashley pump, multiple valves are employed for both suction and delivery, and unusually high pumping speeds are possible with quite moderate water velocities through valves.

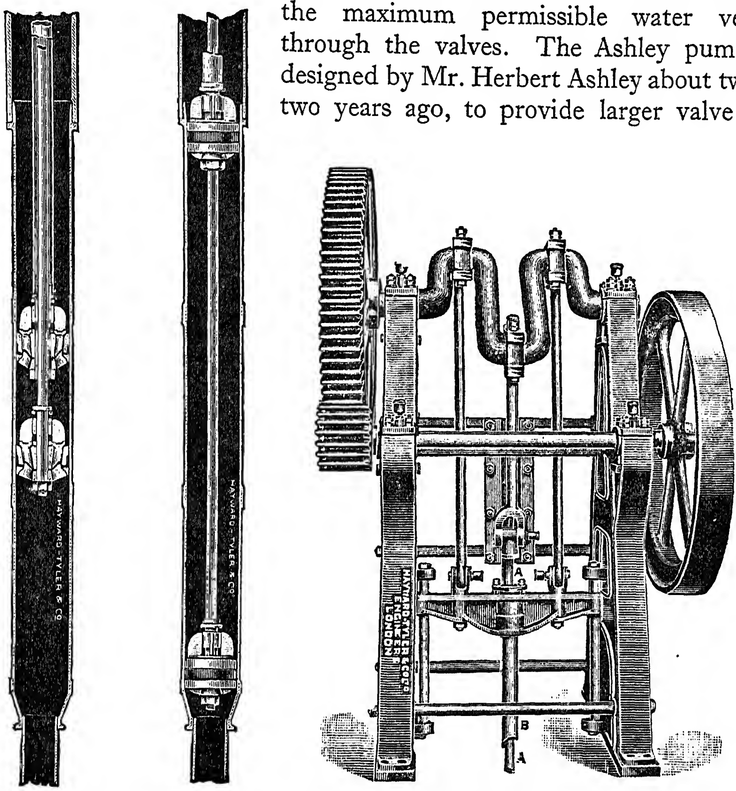


Fig. 13.—Double-acting Tube-well Pump

Many Ashley pumps are successfully working in waterworks, bore-holes, and in mine shafts; the sizes vary from $3\frac{3}{4}$ in. to 23 in. in diameter, and the strokes from 18 in. to 4 ft. long, with mean piston speeds in some cases exceeding 200 ft. per minute.

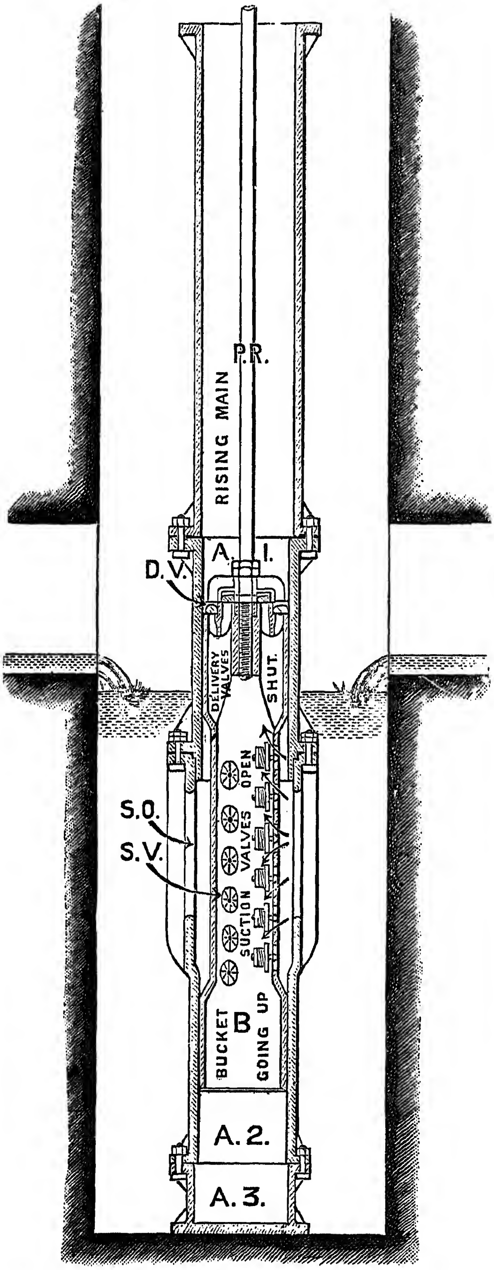


Fig. 14.—Single-acting "Ashley" Pump

Hydraulic Pumps.—With low-pressure pumps it is always the quantity which is the predominant factor in design; free waterways, adequate valve areas, easy changes of direction of flow, and so on, occupy the attention of the designer. With high-pressure pumps, however, it is the head which is the chief consideration, and concentration is made upon the strength of the parts, safety of bearing loads, and the effects of shock due to alternations of stresses.

Hydraulic pumps range from the small hand pumps, frequently used for hydraulic tests in workshops, to powerful pumping engines supplying high-pressure hydraulic service to large cities such as London, Manchester, and Liverpool.

Hydraulic pumps may be belt-driven from a line shaft, electrically driven, or direct driven by steam or internal-combustion engines. An example of a small pump specially designed for hydraulic work and built by Messrs. The Leeds Engineering and Hydraulic Co., Ltd., Leeds, is shown in fig. 15. This has a $\frac{3}{4}$-in. diameter hard steel or bronze ram, 3-in. stroke, and delivers about a third of a gallon a minute against 3000 lb. per square inch pressure. The speed is 80 r.p.m. driven through machine-cut gearing from an electric motor fitted with a hard steel or fibre pinion.

The hollow cast-iron bedplate forms a suction tank from which the pump draws. The valves and seats are of hard bronze, and the ram case and clack box are either of bronze or forged steel.

A typical hydraulic pumping installation built by Messrs. Fullerton, Hodgart, & Barclay, Paisley, is shown in fig. 16. This is a triplex pump with flat guides and single reduction gear, designed to deliver 75 gall. per minute against a pressure of 1500 lb. per square inch, when running at 60 r.p.m.

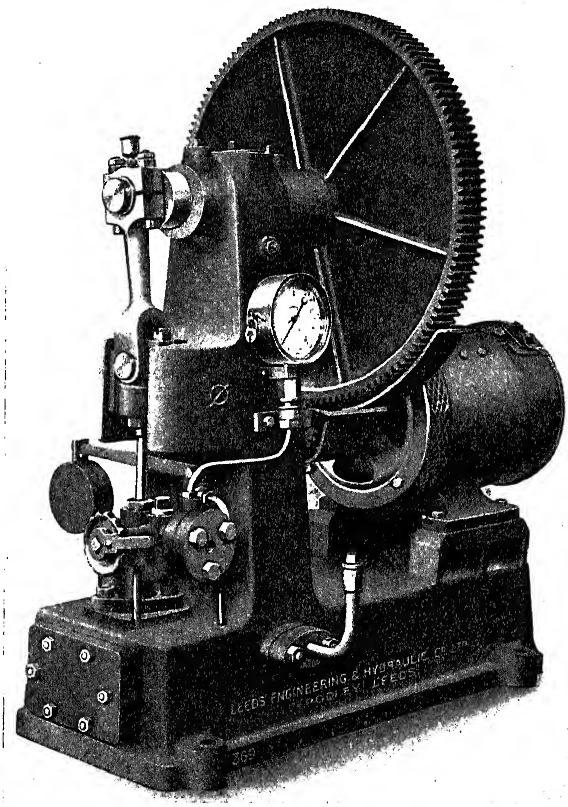


Fig. 15.—Vertical Single-throw Hydraulic Pump

In this case, the plunger diameter being small, the centres of the barrels have been brought close together, and by providing a massive crank-shaft with disc slabs, intermediate bearings have been avoided and a compact pump produced.

For public supply purposes vertical triple-expansion marine-type steam pumping engines are usual, very similar in general design to waterworks pumping engines, to be described later. Typical sets as installed by Messrs. The Hydraulic Engineering Co., Chester, for London, Manchester, and Liverpool may be referred to. These are triple-expansion open-type column engines with surface condensers embodied with the frame, the cylinders

being carried on an entablature cast with the frame. The pumps are bolted vertically to facings on the front side of the condenser, and the piston rods are coupled direct to the pump plungers. Flat guides support the crosshead, back and front, and the crosshead pin is extended to receive the forked ends of the crank-shaft connecting rods. The pump valves are of gun-metal with mitre seats, and are arranged at the bedplate level with curved steel pipes to connect them with the pump. Suction water is supplied by an overhead tank, and the delivery pressure at the station is about 850 lb. per square inch. The rams are $5\frac{3}{4}$ in. diameter and 27 in. stroke, and run from 55 to 60 r.p.m. With steam pressure at 180 lb. per square inch, and a pump horse-power of 267·5, the mechanical efficiency was 85·5 per cent, and steam consumption 12·7 lb. per indicated horse-power.

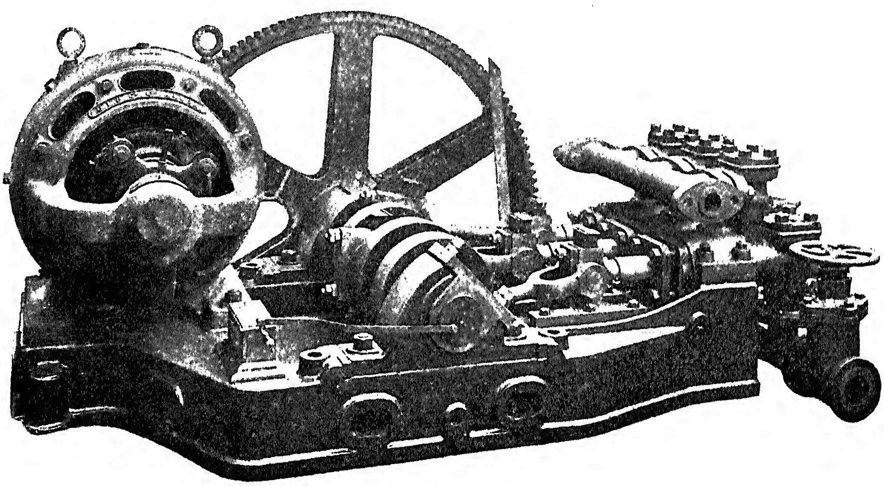


Fig. 16.—Three-throw Hydraulic Pumps

Multi-stage turbine pumps are now being used for hydraulic pressure work, and will be referred to in a later section.

Deep-mine Pumps (reciprocating, power-driven).—One of the severest duties to which pumping machinery is subjected is that of mine drainage. The safe working of the pits is dependent upon the pumps; long runs without a stop are frequently necessary, and the plant is generally handicapped by unavoidable confinement of floor space, long distance from a repair shop, and by surroundings where repairs are unusually irksome.

Now that the electrical equipment of mines has been so extensively developed, the tendency is to install electrically driven pumps, and to distribute them over the most convenient places in the mine, and, if necessary, at different levels in the shaft. This system has important advantages—the pumps may be placed where the water is, at any site in the workings, power is easily brought to the pump with a very moderate transmission loss, and the pumping capacity is readily extended as the mine develops.

The types of reciprocating pumps generally employed are the triplex

and the double-acting duplex (or the quadruplex) forms, each with externally packed rams single-acting in action at each end.

It is now quite common practice to deal with heads of 2000 ft. or so in a single lift, and leading manufacturers have quite a number of their pumps regularly operating against a 3000-ft. head with perfect success.

Owing to the severe duty encountered by these pumps it will be interesting to consider the principal points of design. For high pressures the general arrangement will usually be horizontal, as this gives greater rigidity, and enables the machinery to better withstand the alternations of stress inseparable from a reciprocating pump. Cast-steel valve boxes or " pots " are common practice, and great care is exercised in the arrangement of all waterways, to avoid the occurrence of inertia forces, which would cause pressures in excess of the necessary pumping pressure. The number of reciprocations per minute has an important bearing on the pump stresses, and it will be found that for normal sizes of mine pumps 40 or 50 r.p.m. is usual, and the limiting range rarely goes beyond 30 to 60 r.p.m. The permissible loads on the various moving parts depend upon the surface speeds of the bearings, and upon whether the stresses are alternating or constantly in one direction. For main crank-shaft journals pressures between 400 and 600 lb. per square inch are usual; the crosshead pin 1000 to 1200 lb. per square inch, the crosshead slipper about 40 to 50 lb. per square inch (for cast iron on cast iron), and crank pin pressures of 600 to 800 lb. per square inch are all good practice. The design of the various details for stiffness and rigidity follows well-known methods common to all classes of machinery. The proportioning of the gearing is a matter of very great importance to ensure that only very low tooth pressures are employed, and every care taken to produce as vibrationless a transmission as possible. The valves and seatings are of various types according to the size of the pump, the nature of the water, and the experience of the manufacturer. For small pumps, valves similar to fig. 5, *c*, are used, though often supplemented by a joint ring of leather or cotton-reinforced balata to improve the joint when the water is dirty, and so avoid the very serious scoring through the faces which otherwise occurs with high pressures. A similar idea is very successfully carried out on ring valves in the manner shown at fig. 9, *b*, where the metal portion of the valve carries the weight, and the leather or other jointing material provides the true joint. India-rubber valves are used for light duties on some mine pumps, but are not advisable for heads greater than 250 ft. with ordinary valves, or 500 ft. on rotating valves. Raw hide has been found satisfactory as a seating ring for the highest pressures dealt with. Heavy double-beat valves of the forms shown in fig. 5, *g*, and more frequently fig. 5, *h*, are sometimes fitted into the valve pots of mine pumps; the latter valve when fitted with raw hide or special leather seating rings is very serviceable.

A pump of the double-acting duplex form, built by Messrs. The Prescott Company, of Menominee, Michigan, United States, is shown in fig. 17. These units are built for capacities up to 4000 imperial gallons per minute, and pressures not exceeding 500 ft. head. The plungers are centrally packed,

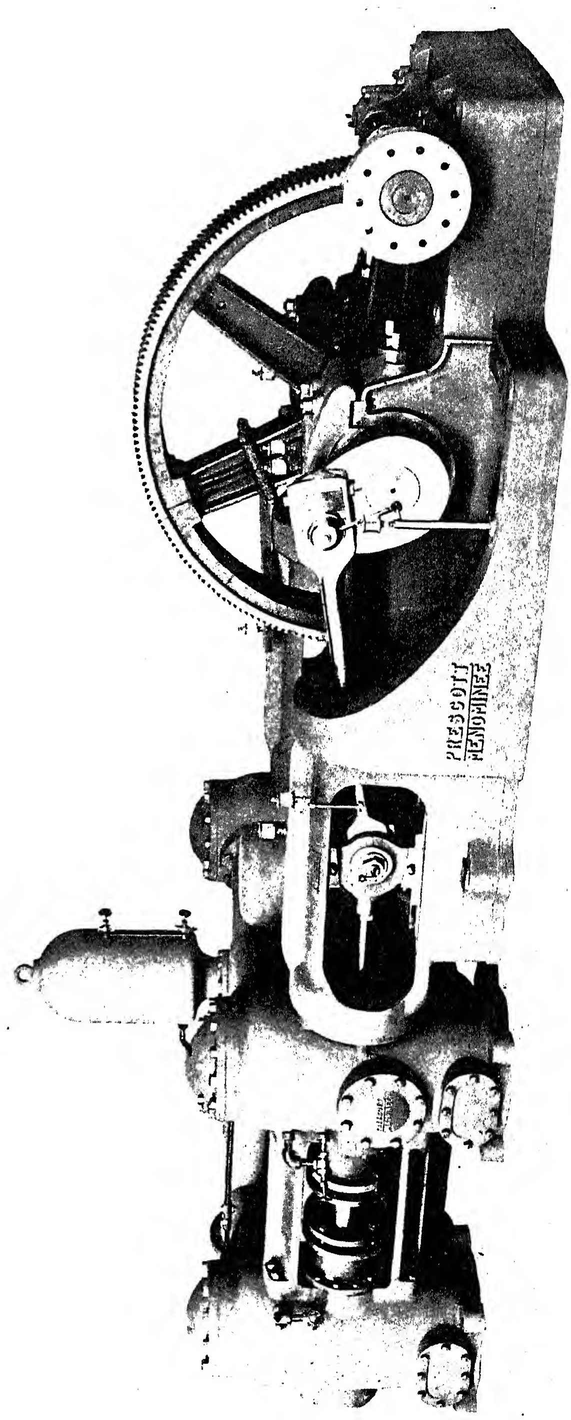


Fig. 17.—Horizontal Double-acting Duplex Mine Pump

and by virtue of the construction, which embodies loose stuffing boxes, glands, and neck bushes as a single unit, they are easily removed for examination or repair. The speed of these pumps is about 52 r.p.m., and this particular design is intended for large volumes against moderate pressures. For higher pressures up to about 1250 ft. head the Prescott Company make a " pot-valve " type of pump of rather more expensive design, and for higher pressures still—say up to 2500 ft.—they build a very massive pump in which the water passages and valve chambers are machined out of solid steel forgings.

Messrs. Glenfield & Kennedy, Ltd., Kilmarnock, have considerable experience of very deep mine pumping, especially metalliferous mines in

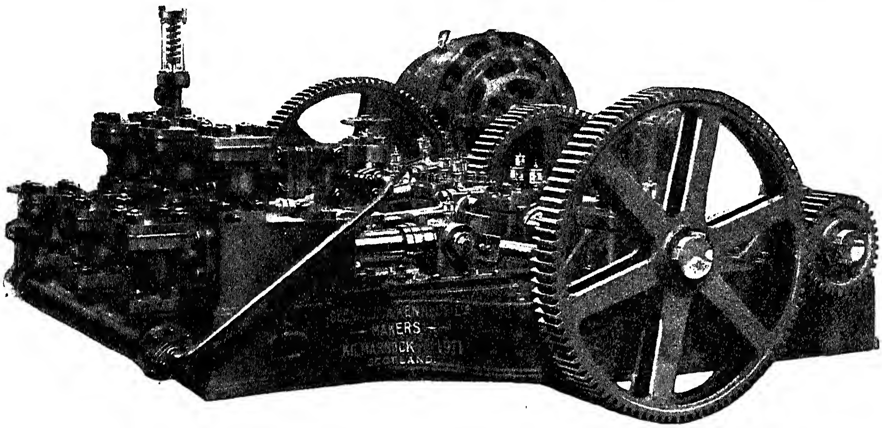


Fig. 18.—High-pressure Horizontal Three-throw Mine Pump

South Africa. A number of their pumps have been operating against heads of over 3000 feet, and are satisfactorily standing the test of several years' continuous work. One of these horizontal triplex pumps is shown in fig. 18, from which the principal features of the construction will be easily gathered.

Direct-acting Steam Pumps (non-rotative).—Pumps which are directly operated by a reciprocating steam-engine—the respective cylinders usually being arranged on the same axis—are called " direct-acting " pumps. On account of the necessity for delivering against a constant water pressure, a single-cylinder direct-acting pump must receive full-pressure steam during the whole of its stroke. Therefore one side of the steam piston is always receiving live steam while the other side is exhausting and becoming chilled in the process. The slower the pump speed the greater the time for cylinder cooling on the exhaust side and consequently the greater the condensation losses and the total steam consumption. It is cylinder condensation which is very largely responsible for the heavy steam consumption inseparable from small direct-acting steam pumps. The advantages of expansive working of steam can be secured, on medium-sized pumps, by fitting two steam cylinders—high pressure and low pressure—arranged either tandem fashion,

or side by side in a duplex pump. In addition to the advantages of expansive working, a compound pump shows a gain in economy due to the reduction of condensation losses brought about by reducing the heat range in each cylinder. Another point is that the higher the steam pressure (temperature) the greater the condensation losses, and it will be found that for steam pressure over 100 lb. per square inch the compound cylinder pump has a more marked advantage over the single-cylinder pump.

General practice in the use of steam pressures is fairly well represented by the following average figures:

For single-cylinder pumps, pressures up to 100 lb. per square inch.
For compound-cylinder pumps, pressures about 120 lb. per square inch.
For triple-cylinder pumps, pressures about 150 lb. per square inch and upwards.

A single-cylinder pump is generally more economical than a duplex (twin high-pressure cylinders) pump because cylinder clearances are halved and steam connecting passages are obviated.

As regards steam consumptions, these may vary very much according to the conditions of operation. Small single-cylinder pumps will be found to consume from 70 lb. of steam per pump horse-power hour in a good pump to 150 lb. or more in an indifferent machine, compound pumps from 33 to 60 lb. of steam per pump horse-power hour, and ordinary triples without compensating gear or fly-wheels from 20 to 30 lb.—large compounds when working condensing rival triples in economy.

For the consumption of direct-acting high-duty pumping engines see Table I (facing p. 58) in the section on waterworks pumping engines

The speeds of direct-driven pumps vary between 50 and 30 double strokes per minute for average sizes, and slower on larger pumps, though in emergency it is common to run pumps quite 50 per cent faster, as in the case of fire-service plant.

The different varieties of direct-acting steam pumps are distinguished by the methods used for operating the steam valves, practically every manufacturer having his own special device, many differing in trifling details only. Taking a broad view of current practice, there are two groups into one of which all constructions will naturally fall.

These are:

1. Those in which the valve is operated mechanically, and
2. Those in which the valve is operated by live steam.

Pumps in the latter category generally, though not always, have a mechanical control of the steam which operates the main steam valve.

The simplest mechanical gear is that of the well-known duplex pump originated by the Worthington Pump Company, and now built by several manufacturers. This uses a common " D " slide valve operated by a rod fitted with adjustable collars; the collars are set so that there is a certain amount of free movement of the rod before the valve is moved. The valve rod itself is moved by the piston rod of the neighbouring cylinder, and the

collars so adjusted that the neighbouring pump is always set in motion before the first pump comes to rest (at about $\frac{5}{8}$ stroke of the other pump is common practice). Thus one piston is always in motion. A typical, though large size, pump of this type, built by Messrs. Hayward, Tyler, & Co., London and Luton, is illustrated in fig. 19, which clearly shows the valve gear.

Well-known examples of pumps with steam-operated valves are the

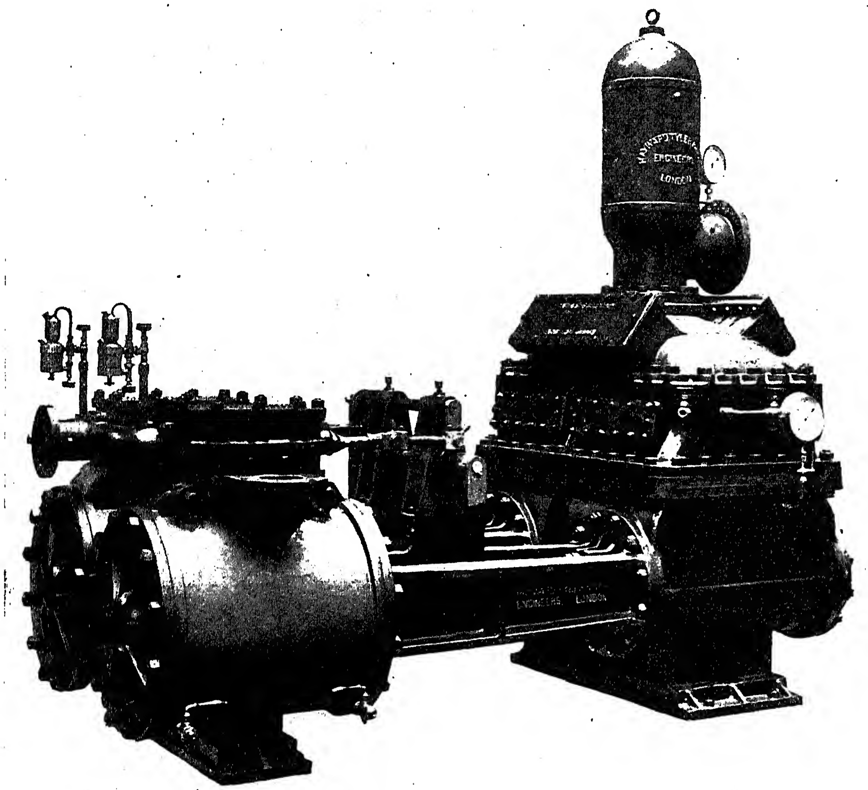


Fig. 19.—Large Size Duplex Direct-acting Steam Pump

Steam cylinder 22 in. diameter. Water barrel 14 in. diameter. Combined stroke 18 in.

Weir pump, Cameron direct-acting pump, the Warren steam pump, the Davidson pump, the Hall pump, Tangye, Blake, and many others.

All the types mentioned employ a mechanical gear to operate an auxiliary valve which controls the steam supply to a main valve—generally of the shuttle type—which in turn slides across the main cylinder ports and controls the movement of the pump piston. A typical example of a steam-controlled valve without " tappets " or mechanical gear is the Evans " Cornish " steam pump. The steam cylinder in this case is provided with ports, opened and closed by the main piston, and communicating with a shuttle valve which is operated by the steam so admitted.

The fundamental idea underlying all these gears can be grasped by the examination of one of them. Fig. 20 shows a sectional drawing of a " Davidson " valve gear as built by Messrs. M. T. Davidson Co., Brooklyn,

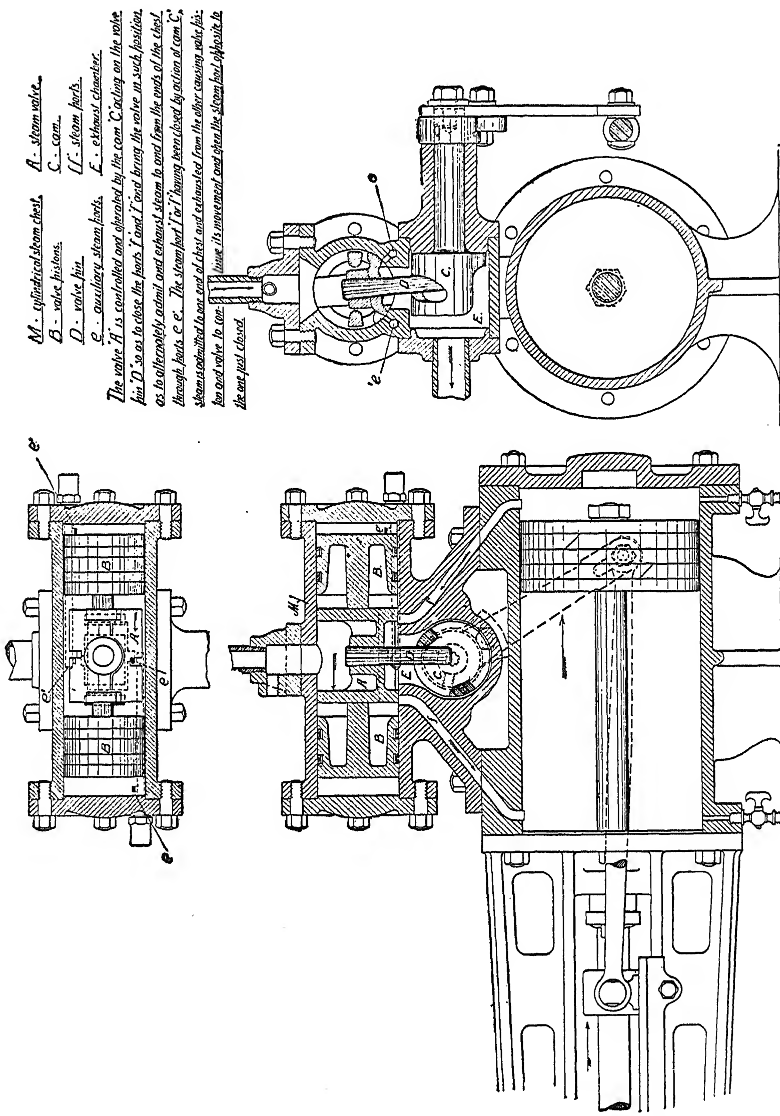


Fig. 20.—" Davidson " Valve Gear for Steam Pump

New York, and by Messrs. Sir W. H. Bailey & Co., Ltd., Salford, Manchester.

The valve gear consists of a valve, valve pistons, valve pin, and cam.

The valve is controlled and operated by the steel cam C, itself connected with the main piston rod, acting on the steel pin D, which passes through the valve into the exhaust port in which the cam is located. In addition

to this mechanical operation, steam is alternately admitted to and exhausted from the ends of steam chest by ports *e* and *e'*, operating the pistons B and B'.

The pump being at rest, with the valve A covering the main steam ports *f* and *f'*, the cam C holds the valve by means of valve pin D, so that ports *e* and *e'* admit steam to one end of chest and connect the other end with exhaust port; the steam acting on valve pistons will move valve pistons and valve, opening main ports *f* and *f'*, admitting steam to one end of the steam cylinder and opening the other end to the exhaust.

Steam being admitted to cylinder by one of the main ports, as *f* in illustration, the steam piston, cam, valve, &c., will move in direction indicated by arrows. The first movement of the cam will be to oscillate the valve, preparatory to bringing it into proper position for the opening of the auxiliary steam port *e* to live steam and *e'* to exhaust, and secondly, to bring the valve to its closure (mechanically) slightly before the end of the stroke of main piston (thereby causing slight cut-off and compression), and fully opening auxiliary port *e* to steam and *e'* to exhaust. By the admission of steam to one end of chest, the other being open to exhaust, the valve pistons will move valve to such position as will allow the admission and exhaust of steam to and from cylinder for the return stroke.

With regard to the water end of direct-acting steam pumps, the design varies according to the pumping duty on the lines already indicated for various requirements. The ordinary trade pump is generally fitted with multiple valves of the types (fig. 5, *a* and *b*) already described, and larger pumps are fitted with pot valves or any of the other types indicated for such services as are demanded.

Boiler-feed Pumps.—The reciprocating boiler-feed pump is simply a small size direct-acting steam pump, and all the types just referred to are used for boiler feeding. Certain requirements affecting reliability, proportions of steam-piston diameters to plunger diameters, materials to deal with hot water, &c., are called for with boiler-feed pumps, and therefore many firms make a distinct line for this purpose quite apart from their ordinary trade pumps.

A conspicuously efficient boiler-feed pump is that made by Messrs. G. & J. Weir, Ltd., Glasgow, and shown in section in fig. 21. The steam valve is of the shuttle type, controlled by an auxiliary valve which is itself actuated by levers connected with the common piston rod. The water valves are of the Weir group type, made of bronze and with Admiralty gun-metal seats. The pump rod is of cold rolled manganese bronze, and the plunger of gun-metal fitted with special ebonite packing. Other forms of this pump include tandem compound and twin compound designs, both of which naturally give a greater steam economy than the single-cylinder pump. The special Weir features have also been embodied in designs of pumps for general purposes and hydraulic pressure service.

Waterworks and Drainage Pumps.—For general waterworks and public supply purposes, and for town drainage, all forms of pump are used, according to the special conditions at the site, the volume and water pressure

required, and according to the expenditure permissible. The term " waterworks pumping engine ", however, stands for a distinct type which is the lineal descendant of the original Newcomen atmospheric pumping engine, improved by James Watt.

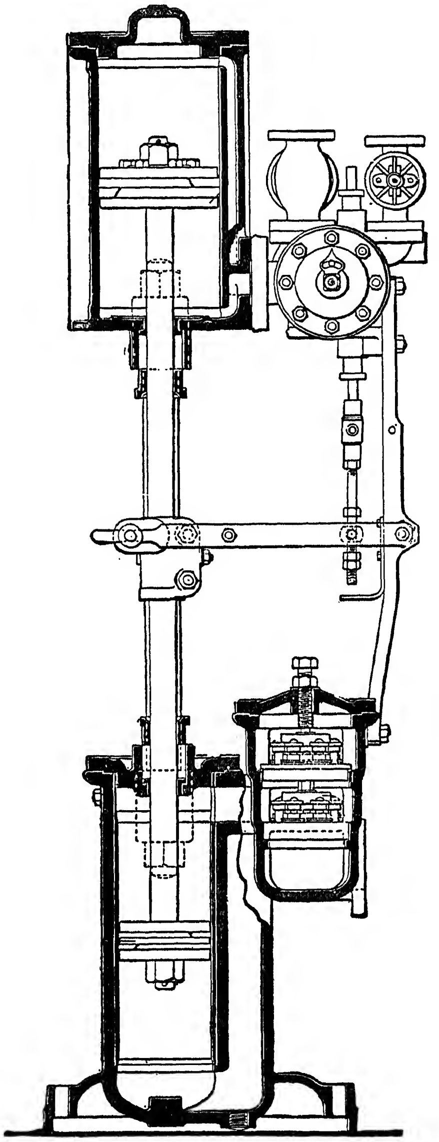


Fig. 21.—Section of Weir Boiler-feed Pump

Waterworks pumping engines may be conveniently classified into:

1. Rotative class, including
   (*a*) Beam engines—with and without fly-wheels, and
   (*b*) Crank and fly-wheel engines.
2. Non-rotative or direct-acting class, including
   (*a*) Simple forms, and
   (*b*) Differential or compensated types.

Both horizontal and vertical forms exist of the last three varieties.

To deal adequately with waterworks pumping engines would require a treatise apart, and it is only possible therefore to select a very few typical machines for special notice.

Duty.—In comparing the behaviour of pumping engines it is usual to refer to their " duty ", and this term has had various interpretations at different periods. At the present time duty is generally based upon the number of foot-pounds of work done, and measured by the water raised for the consumption of 1000 lb. of dry steam, or upon 1,000,000 B.Th.U. used by the pumping engine.

The term " high duty " now generally used for modern pumping engines simply means " high economy ", or a high numerical value for the duty as calculated on either of the above bases.

Rotative Class.—The Cornish beam pumping engine is represented in a very large number of town and city pumping stations where it was installed during the period 1836 to about 1885.

This form of pumping engine, however, is no longer made, and in the few cases where the beam is still retained, a heavy fly-wheel is driven from a connecting rod and crank connected with one end of the beam.

In the crank and fly-wheel pumping engine the beam has disappeared altogether, and the cylinders are in direct line, horizontally or vertically, with the crank-shaft. There is a very great variety in the arrangement of this class of engine, the arrangement often being influenced by the individual taste of the designer, and very largely varied by the differences in sites, availability of the water, and the pumping requirements. These engines do not lend themselves to standardization, and must generally be modified to suit the job, and nearly always require to conform to the very latest improvements of the day. A typical modern engine of this type would generally be made with steam-jacketed cylinders—compound for quantities up to one or two million gallons a day, and triple for larger sizes—using steam at 120 to 180 lb. pressure, moderately superheated, the receiver steam also reheated, and Corliss valve gear would be usual. The piston and plunger speed would be 200 to 250 ft. per minute, and the pumps would be single-acting if vertical and double-acting if horizontal. The duty of the unit would average 110 to 130 million foot-pounds, in water raised, per 1000 lb. of dry steam, for ordinary compounds of capacities up to about a million gallons a day; and, for triples, of capacities of 2 to 3 million gallons a day upwards, the duty would vary from 150 to 170 million foot-pounds, according to size, steam pressure, superheat, and other conditions. Departures from this specification would, not uncommonly, occur.

The installation recently completed by Messrs. Hathorn, Davey, & Co., Ltd., Leeds, for the Margate Town Council forms a good example of a modern plant of moderate size. The work to be done was to deliver 4 million gallons per twenty-four hours against a total head of 430 ft. The water is drawn from a well, and in the first instance has to be lifted 130 ft. by well pumps; then at the surface it is taken up by the force pump and forced 15 miles against a head of 300 ft. including pipe friction. The engine is a vertical triple Corliss, fly-wheel type, and operates three vertical plunger pumps, placed below the steam cylinders, direct from the crossheads, and a three-throw well pump from an extension of the crank-shaft, thus avoiding all gearing.

The well pumps are fitted with the firm's special type of buckets and clacks. These are of multi-annular design with a series of concentric ring valves (seven in the Margate pump) stepped one above the other to produce a conical-headed bucket. By this means a full waterway is secured with a very small valve lift. The features of this valve will be appreciated by contrasting it with the more usual form of double-beat bucket valve shown at *i* in fig. 5. There are more working parts, but quieter action should be secured with the multi-valve type.

In November and December, 1920, two trials were taken on the Margate pumping engines, with very similar results except that the superheat was greater in one case than the other, and the economy is

influenced accordingly. A summary of the figures obtained is as follows:

	Trial 5th Nov., 1920	Trial 21st Dec., 1920
Duration of trials (hours)	12	6
Steam pressure at engine stop valve (lb.)	161·7	161·5
Superheat at engine stop valve (deg. F.)	10	34
Average indicated horse-power	260	267
Average pump horse-power	241·49	242·69
Over-all mechanical efficiency, engines and pumps (per cent)	92·6	90·9
Total steam used per hour (lb.)	2888	2743
Steam used per indicated horse-power hour (lb.) ..	11·1	10·3
Steam used per pump horse-power hour (lb.)	11·96	11·32
Or, allowing 5 per cent for loss of power by slip (lb.) ..	12·56	11·89
Equivalent duty per 1000 lb. of steam (million ft.-lb.) ..	157	166

The leading particulars of the unit are given in Table I.

Generally speaking, on this class of engine, whether used for raising water from wells or for forcing water already at the surface, 1 horse-power in water lifted can be obtained for an expenditure of 11 to 12 lb. of steam per hour.

A general-arrangement drawing of the pumping engine showing the auxiliary plant is given in fig. 22.

A very large American pumping engine of the crank and fly-wheel type is shown in fig. 23. This equipment is capable of pumping 25 million imperial gallons per twenty-four hours, and was built to the order of the Louisville Water Company, Louisville, Kentucky, United States, by the Allis-Chalmers Manufacturing Company, Milwaukee, United States.

American waterworks pumping engines are frequently of much larger sizes than those built in this country, for two reasons: (1) a larger allowance of water per head of population is generally required, and (2) the practice of scattering small pumping units throughout populated areas is not followed to the same degree as in this country. Larger pumping engines should naturally give better economies proportionate to the water raised.

Non-rotative or Direct-acting Class.—Single-cylinder direct-acting steam pumps use steam too extravagantly to suit waterworks requirements, and the compound forms are generally too wasteful for anything but very small plants. Direct-acting forms are now generally of the triple-expansion type arranged either horizontally or vertically. To use steam economically it must be used expansively, which entails a varying pressure on the steam piston. The pressure on the water plunger, however, is constant throughout the stroke, therefore some method of equalizing the work done by the steam throughout the stroke is necessary. Quite a number of ways of doing this have been actually put in service, the best-known of which is the Worthington device of oscillating compensating cylinders, which store power during the initial pressure portion of the stroke, and give out power during

Table I.—PERFORMANCES OF WATERWORKS PUMPING ENGINES

Year.	NAME. Site.	NAME. Maker.	REQUIREMENTS. Millions of Imperial Gallons per 24 hr.	REQUIREMENTS. Head in Feet.	SIZE. Steam Cylinders, Inches.	SIZE. Pump Barrel, Inches.	SIZE. Stroke, Inches.	PARTICULARS. I.H.P.	PARTICULARS. W.H.P.	PARTICULARS. Steam Pressure, Lb./Sq. In.	PARTICULARS. Superheat, Deg. F.	PARTICULARS. Vacuum, Inches.	PARTICULARS. R.P.M	PERFORMANCE. Mechanical Efficiency, Per Cent.	PERFORMANCE. Pounds of Steam per I.H.P. Hour.	PERFORMANCE. Pounds of Steam per W.H.P. Hour.	PERFORMANCE. Millions of Ft.-Lb. of Work Done. Per 1000 lb. Steam.	PERFORMANCE. Millions of Ft.-Lb. of Work Done. Per Million B.T.U.	PERFORMANCE. Thermal Efficiency.
..		Worthington-Simpson	2·5		12 + 20 + 34	13½	24	125/130	—	—	150°	—	—	—	—	12·2	—	—	—
1900	Leeds	Hathorn Davey	2·77		15 + 25 + 40	13½ 3 rams	36	183·7	167·6	138	None	27·8	34·6	91·3	11·91	13·4	152	138·6	16·9
	Bristol	Hathorn Davey			16 + 28 + 46	18 3 rams	36	184·6	167·9	160	—	—	27·5	91	—	12·58	—	—	17·96
1905	Rosario	Hathorn Davey	4	H.L.: 170 L.L.: 55	15 + 25 + 40	High lift: 16 Low lift: 16¾	36	232	209	177	53°	28	36	90	11·1	12·6	161	147	18
	Toronto	Inglis	6·04	172	16 + 30 + 44	21⅝	36	—	—	163	—	21·2	29·3	—	—	12·12	163·4	150·2	—
	Hong-Kong ..	Worthington-Simpson	3	287 to 387	18 + 31 + 50	15⅜	36	—	—	150	120°	—	30	90[1]	—	11·5[1]	172[1]	—	—
1920	Margate	Hathorn Davey	4	Well: 130 Force: 300	20 + 36 + 53	Well: 16 Force: 15½	48	267	—	161·5	34°	—	25·7	90·9	10·3	11·32	166	—	18·6
1913	Lincoln	Ashton Frost	3	Well: 102 Force: 423	26 + 45 + 68 48-in. stroke	Well: 18 Force: 16¼	60 48	385	345	170	163°	28·2	20	91·3	Mean 9·82	Mean 11·15	Mean 177·8	—	18·7
	Odessa	Hathorn Davey			20 + 36 + 54	—	42	417	380	180	5°	27·75	38	91	11·35	12·8	—	—	17·6
1910	Widnes,[2] Stocks Well Ext.	Hathorn Davey			23 + 38 + 60	2-Bore: 20 1-Force: 20	72	491	410·6	175·5	81·5°	—	17·14	83·6	11·42	13·8	143·47	—	—
1910	Rand	Hathorn Davey	2	900 to 960	23 + 43 + 64	12 3 rams	36	502	461	182	120°	—	37·5	91·8	10·37	11·33	173·6	—	18·8
1895	Boston,[3] Chestnut Hill	E. D. Leavitt, U.S.A.	17·5	137	13·7 + 24·3 + 39 72-in. stroke	3 rams: 17½ 48-in. stroke Double acting	72	565	504	185	—	27	50·6	89·4	11·22	12·54	158·1	—	—
1893	Milwaukee, North Point	Ed. P. Allis, U.S.A.	14·4	161·8	28 + 48 + 74	32 3 rams	60	574	520	125	None	27	20	90·6	11·8	13·3	154	137	17
	Rangoon	Worthington-Simpson	14·4	200	16 + 26 + 50	25½	48	675	607	200	120°	28	27·5	90	—	[4]	—	—	—
	Indianapolis ..	Snow, U.S.A.	16·7		25 + 52 + 80	—	60	775	—	170	—	25	21·5	95·4	11·26	—	167·8	150·1	—
	Palermo	Worthington-Simpson	14·75	230	23 + 36 + 66	31¼ Double acting	60	800	714	200	100–150°	28	16·5	—	11·5[5]	—	—	—	—
—	Boston	Allis-Chalmers, U.S.A.	25		30 + 56 + 87	—	66	—	—	185	—	—	17·7	93·1	10·3	11·09	178·4	163·9	21·63
1906	Bissets Point ..	Allis-Chalmers, U.S.A.	17·2	100	34 + 62 + 94	33⅞	72	865	—	158	—	27·3	16·54	97·4	10·6	—	181	159	—
—	Buffalo	Lake Erie Engineering Co.	25	198	37 + 63 + 94	42	60	1185	—	182	—	25·5	20·8	94·9	12·39	—	152	135	—

[1] Guarantee figures given on 350 ft. mean head.
[2] Horizontal Corliss gear engine.
[3] Leavitt vertical inverted engine. Diagonal pump—Riedler valves
[4] Guarantee: 2¼ lb. of Indian coal per w.h.p.-hour.

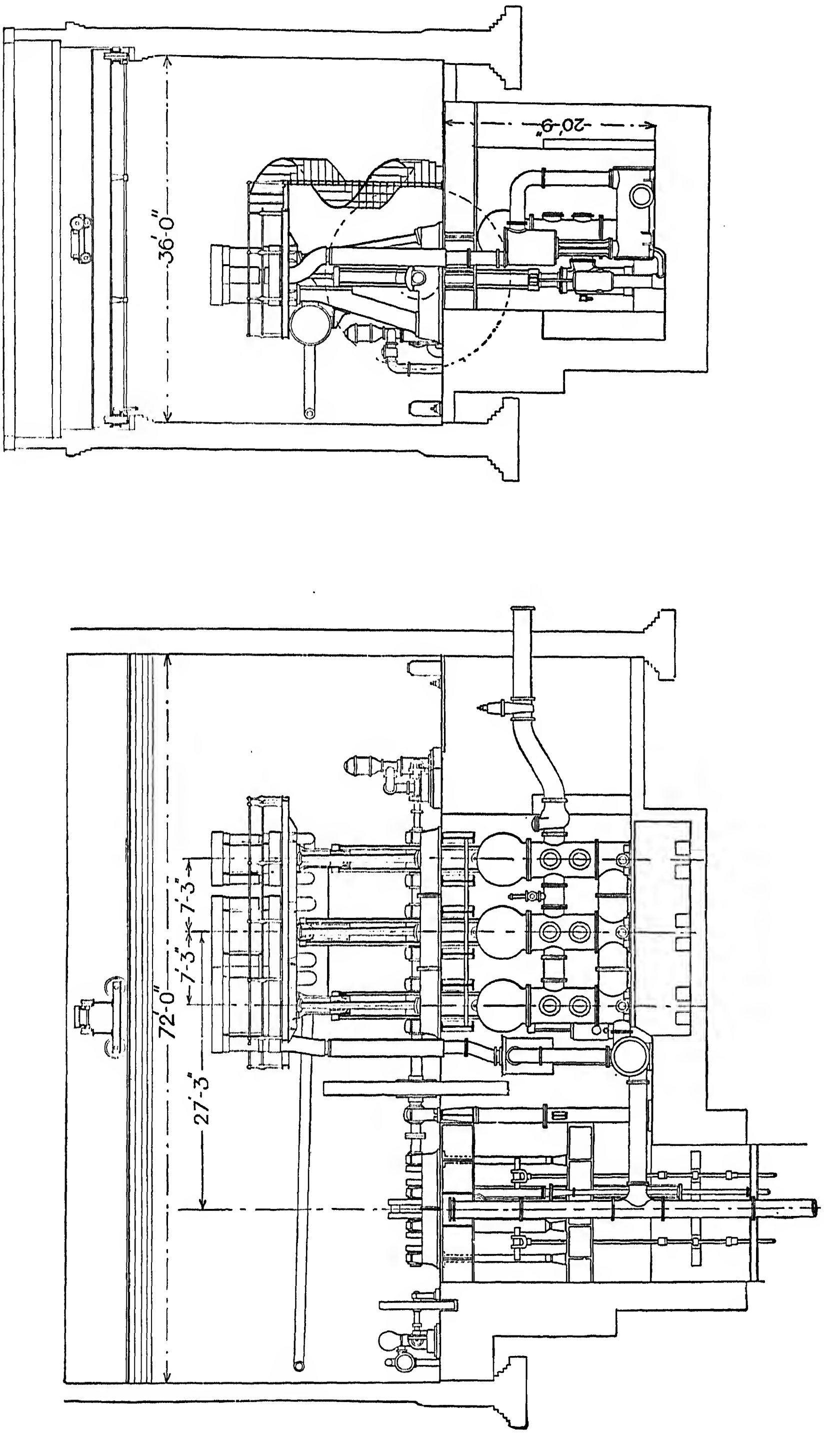


Fig. 22.—General Arrangement of Pumping Plant at Margate Waterworks

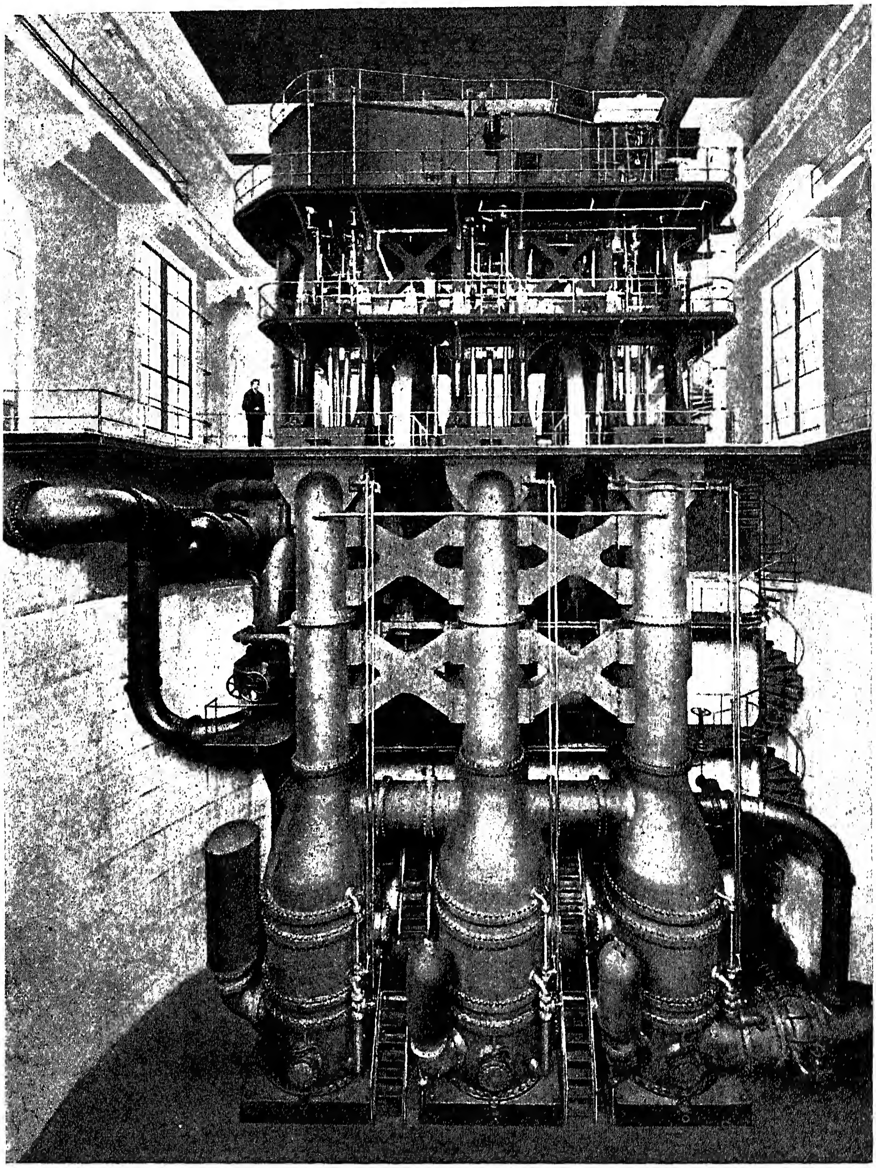


Fig. 23.—25-Million-gallon Pumping Engine built for Louisville Water Company, Louisville, Kentucky, by the Allis-Chalmers Manufacturing Company

the expansion portion of the stroke. Modern examples of the Worthington type of triple-expansion direct-acting pumping engines are a batch of twelve vertical sets for Buenos Aires Water Supply at Palermo (Argentine),* particulars of which are given in Table I, and similar but horizontal engines for

* A photograph and reference to these engines was given in the *Proceedings of the Institution of Mechanical Engineers* for December, 1914, p. 811, and Plate No. 23.

Rangoon Waterworks, India, also listed on the same table, and illustrated in fig. 24.

An idea of the performances of modern waterworks pumping engines is given by the figures on the accompanying Table I (facing p. 58).

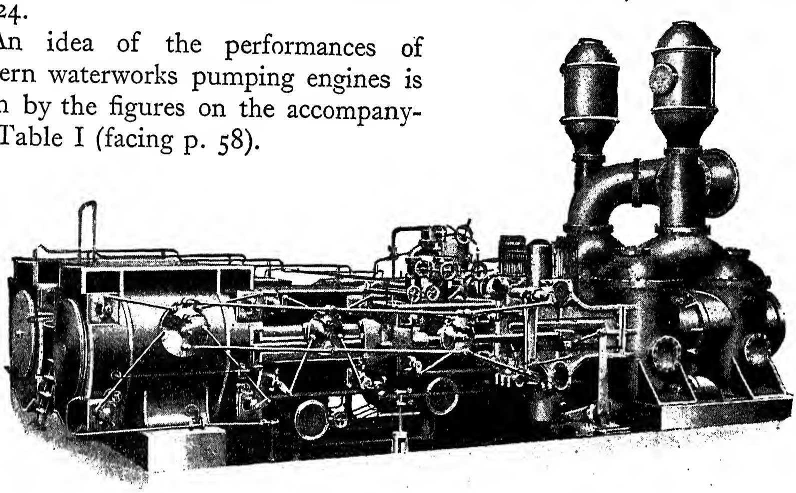


Fig. 24.—Single-barrel Pump on Column

ROTARY PUMPS

Centrifugal and Turbine Pumps.—The centrifugal pump in its simplest form consists of three parts (see fig. 26): (1) casing, (2) impeller, and (3) spindle.

The casing or housing is provided with inlet (suction) and outlet (discharge) branches, and with a stuffing box to permit of the projection of the spindle through the casing. The function of the casing is (*a*) to enshroud the impeller, (*b*) to control the water at its entrance to the impeller, and (*c*) to collect the delivery water from the periphery of the impeller.

The impeller is a hollow wheel provided with arms, or vanes, and its duty is, when revolved, to communicate energy to the water which has to be pumped.

The spindle is the means of transmission of motion and energy from the driving motor or pulley to the impeller.

When the impeller is rotated the body of water contained therein is also revolved, and naturally leaves the centre and travels outwards. As a result there is a tendency to form a vacuum at the centre of the impeller, and water is drawn in through the inlet branch to take the place of the water thrown out from the periphery of the impeller into the outlet branch. Thus pumping is maintained continuously. By virtue of its rotation an impeller creates a pressure difference between the two bodies of water, respectively at its periphery and at its eye (or centre), regardless of what the absolute pressures may be at either of these points. For this reason it is possible to group

several impellers together in series on one spindle, and to pass the water through each of the impellers in turn and so obtain greater and greater pressure according to the number of impellers (or stages) employed. It is on this principle that multi-stage centrifugal or turbine pumps are constructed. The pressure difference created by an impeller, or by each of a series of impellers, is a function of the tangential velocity of the water at the impeller circumference, or for present purposes we may consider it to be a function of the rim speed of the impeller itself. The relation between this rim speed V and the pressure difference H expressed in "head" units, is such that

$$H = K\frac{V^2}{2g}$$

where K is a constant wnose value closely approximates to unity, and g, of course, is the acceleration due to gravity.

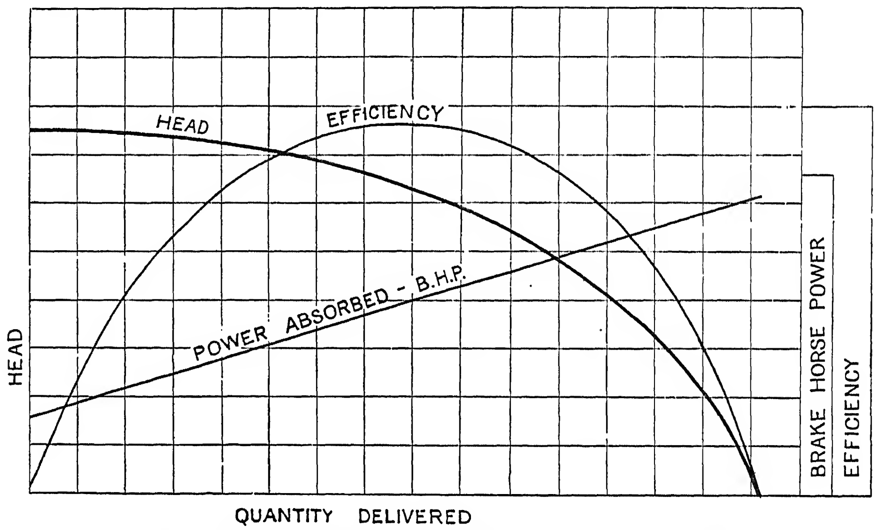


Fig. 25.—Characteristic Behaviour of a Centrifugal Pump at Constant Speed

When K has unit value this formula is seen to be identical with the well-known dynamic formula for the greatest height attained by a body thrown vertically upwards.

Centrifugal pumps are generally driven at a more or less constant rotary speed, and it is therefore necessary to examine their behaviour under this condition. Every different design of pump will give a slightly different characteristic behaviour, but fig. 25 gives the simplest possible typical shape of behaviour curves for a pump running at a constant speed. The diagram is plotted on a "quantity" abscissa, and the ordinates represent "head", "power" (b.h.p.), or "efficiency" according to the graph under consideration. Examining the head curve, it is seen that with zero delivery, i.e. running with a closed outlet pipe, a certain pressure head is created which bears the

relation already given to the rim speed of the impeller. If, however, the pump is discharging water it is seen from the curve that the head against which it can deliver, at the given speed, becomes less as the quantity becomes greater. Considering the " brake horse-power " line it is evident that the power consumed is a minimum when no water is being delivered, and when the pump is only creating pressure head. As the quantity increases the necessary power increases, and becomes a maximum when the quantity is a maximum.* It should be noted that the head falls as the brake horse-power increases, and in this respect the centrifugal pump, operating at constant speed, differs from the plunger pump. The reason is quite obvious. The power absorbed by a centrifugal pump is proportional to the momentum it imparts to the water, and the greater the quantity of water passing the pump the greater is the momentum imparted; therefore, generally speaking, the brake horse-power absorbed is greater also. Referring to the effi-

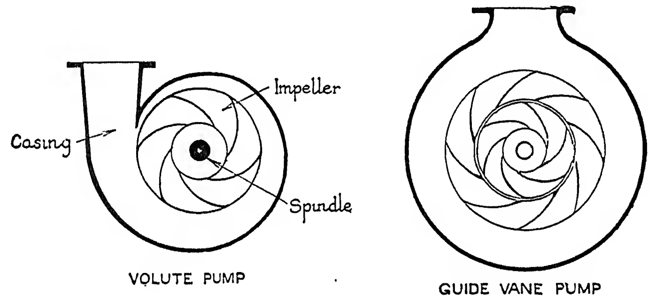


Fig. 26

ciency curve, it is seen to rise from zero to a maximum and fall to zero again. That is to say, there is a certain range of conditions for every centrifugal pump within which it operates at its best efficiency. This best efficiency range is different for every speed, and bears a definite relation to that speed.

Various Types.—There are two main groups of centrifugal pumps,

(*a*) Volute pumps. (Fig. 26.)
(*b*) Guide-vane pumps. (Fig. 26.)

The volute is a collecting chamber or spiral housing disposed around the impeller, and designed with a gradually increasing area to suit the volume delivered by the impeller. In some cases a simple diffuser chamber without vanes is fitted between the impeller and the volute. Guide-vane pumps have diffusion vanes between the impeller and the collecting chamber, and occasionally guide vanes are combined with a spiral housing in a volute

* In all modern pumps the brake horse-power line either flattens or bends downwards soon after the maximum efficiency point is passed. The diagram is purposely drawn to show the simplest possible case. Flattening of the brake horse-power line is attributed to reaction on the impeller when the outgoing water attains a high enough velocity. This reactive effect can be obtained to any desired degree, and is to be found in varying amounts in all modern designs.

pump. As a rule volute pumps are used for low and moderate pressures, or where large volumes of water are being pumped. The introduction of guide vanes adds expense to the pump, and they are generally only fitted for higher heads or where the volume to be delivered is very small in comparison with the head. Different types of pump may be distinguished by the way in which the impellers and casings are constructed. The principal common varieties of impellers are shown in fig. 27, and the common forms of casings and combinations in figs. 28 and 29.

Any of these varieties may be arranged either with the spindle vertical

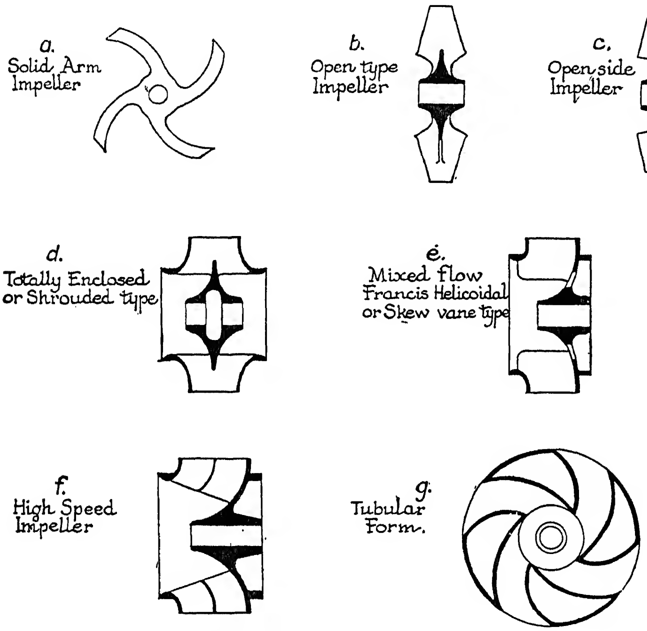


Fig. 27.—Principal Varieties of Impellers

or horizontal. The solid arm impeller, fig. 27, *a*, is used on cheap pumps, and for comparatively low heads. Type *b*, fig. 27, is used on large and small pumps for low heads, and when well designed will give remarkably good results. The open-sided impeller *c* is common on very small cylinder jacket circulating pumps on petrol motors, and in larger sizes is chiefly met in certain American pumps. Type *d* is generally recognized as the standard design for a high-efficiency low-, medium-, and high-lift pump, and is extensively used either as a single inlet (fig. 28, *a*) or double inlet impeller. "Francis type" impellers *e* are used when it is desirable to keep the size of the impeller to the smallest possible dimensions. High-speed impellers *f*, with or without a mid-rib, are quite a modern development introduced to enable high rotary speeds to be adopted on low heads. They are made in single- and double-flow form. Tubular impellers *g* in various modified forms are used on pumps intended to deal with solid material in suspension in the water

pumped. The object is to provide full-way passages throughout without any constrictions or convergent passages.

As regards casings, fig. 28, *a*, represents a good inexpensive form for low heads, and particularly for small and average size pumps, type *b* similarly for larger sizes. The horizontally divided casing *c* is common for low pressures, and chiefly for large pumps, as it provides convenient access to the interior of the pump with the minimum amount of dismantling. This type is also very widely used in American practice down to quite small

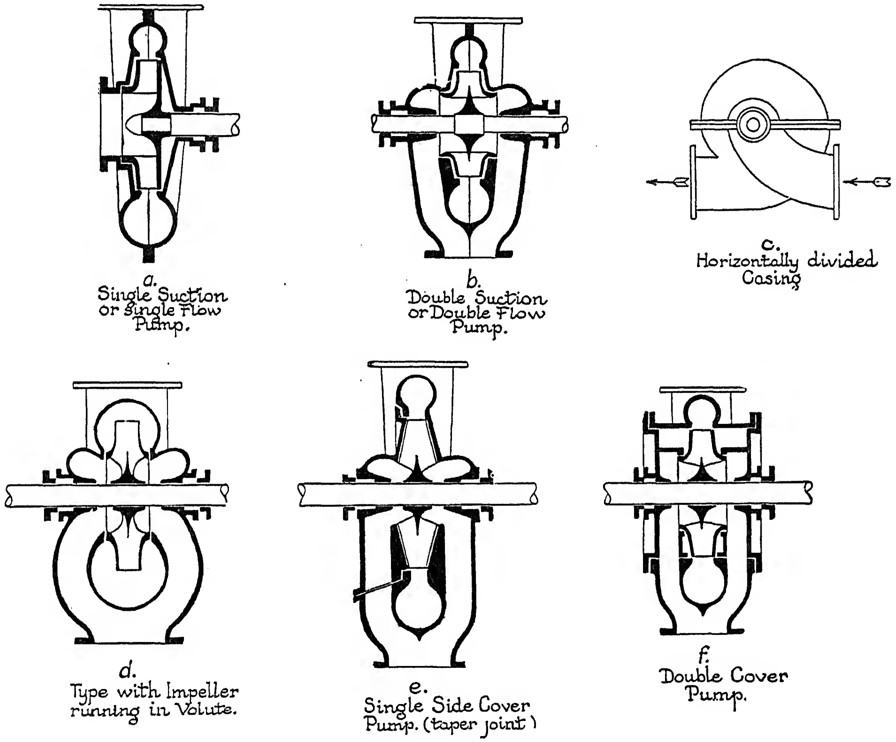


Fig. 28.—Common Forms of Centrifugal Pump Casings

pumps. Type *d* is generally made with a horizontal division of the casing, and is extensively adopted for ship's circulating pumps, dock pumps, &c. Type *e* is the original Gwynne "Invincible" pattern, and type *f* an increasingly popular form suitable for a very wide range of duties. Typical forms of "series" and "parallel" pumps are shown in fig. 29, and also a completely lined pump. The series or multi-stage pump is the high-pressure form of the centrifugal pump, and is extensively used for mine drainage, fire extinction, waterworks, hydraulic pressure systems, and, in fact, every form of pumping against high and moderate pressures. The parallel pump is generally coupled to a steam turbine or electric motor, and enables a high rotary speed to be used when pumping against a low pressure.

Classification of Centrifugal Pumps.—Centrifugal pumps for general work are classified according to the pressures against which they deliver.

1. Very low lift pumps, heads from 0 to about 20 ft.
2. Low lift pumps, heads up to 80 ft.
3. Medium lift pumps, heads up to 150 ft.
4. High lift pumps, heads up to 4000 ft.

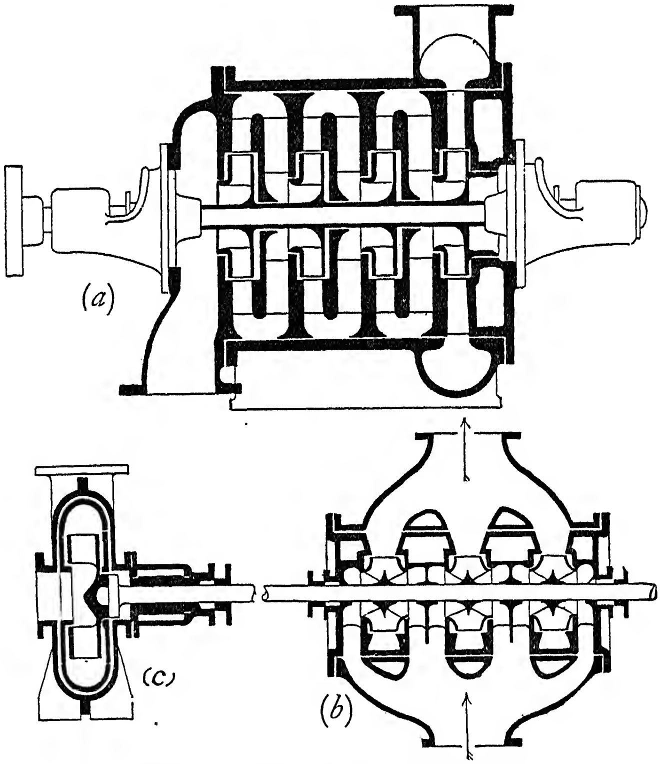


Fig. 29.—Combination Pumps and Special Form

Manufacturers' practice differs chiefly concerning medium lift pumps. Some makers supply medium lift pumps of the volute type for heads up to 200 ft. or higher, while others provide a single-stage guide-vane pump for such duties. The choice between the two is principally one of cost, though if the desired quantity of water is small, a higher efficiency will be secured from the guide-vane type. With a sufficiently large delivery of water a volute form of pump can compete with a guide-vane pump in efficiency as a single-stage unit to heads as high as 500 ft. Needless to say it is an exceedingly rare occurrence to meet with such pumping duties.

Accessories.—The equipment of a centrifugal pump is much the same as a reciprocating pump except that air vessels are unnecessary, and some form of charging apparatus is indispensable. Fig. 30 shows typical

accessories for a pumping plant, and is self-explanatory. The filling funnel is a very common device for charging a centrifugal pump, although many other arrangements are in use, dependent upon the conveniences at hand, and the volume of pump and piping that has to be filled. When steam is available an air-exhausting steam ejector is often fitted, and at other times a vacuum pump. For small centrifugal pumps the vacuum pump would be hand operated, and for larger sizes a power-driven pump would be necessary.

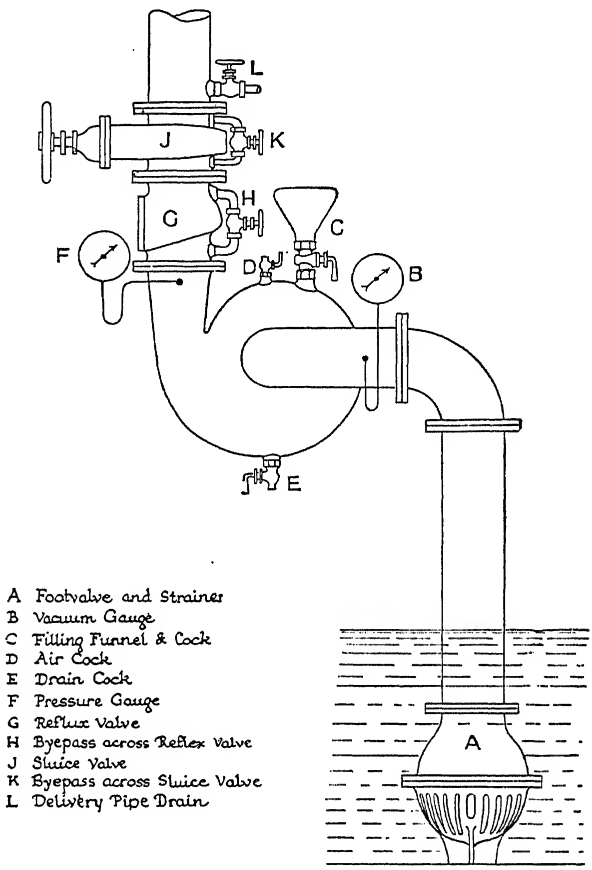


Fig. 30.—Centrifugal Pump Accessories

Centrifugal Pump Practice

Very Low Lift Pumps.—To obtain a high efficiency on heads below 10 ft., and without producing a pump of inordinately large dimensions for the desired quantity, is probably one of the most difficult problems which besets a designer. An interesting solution of this problem is supplied by Messrs. Ruston & Hornsby, Ltd., of Lincoln, who manufacture a specially low lift pump for dealing with lifts of 1, 2, and 3 metres head, for irrigation work in Egypt. The mechanical construction of the pump is similar to fig. 28, *b*. Such pumps are generally driven by belt from oil engines, but may be direct coupled to electric motors or driven in any other convenient manner.

Open-type pumps are sometimes used where very large volumes of water have to be raised a few feet only, and in these cases it is not usual to make casings at all, or should a casing be desirable it is swept out in simple form in concrete. Fig. 31 shows an open-type pump constructed by the Geue-Pumpenbau Gesellschaft, Berlin. Here the lift varies between 1 ft. and 3·6 ft., and the capacity is between 5000 and 6000 gall. per minute. The motor is about $8\frac{1}{4}$ b.h.p., and runs at 965 r.p.m., driving the pump at 200 r.p.m. through a worm gear. The impeller design is similar to fig. 27, *e*,

and simple guide vanes are introduced to reduce the shock losses at exit from impeller.

Low-lift Pumps.—The great majority of centrifugal pumps that are built fall under the category of low-lift pumps, and as far as the principles of design are concerned this term covers many of the special forms of pump such as " marine pumps ", " sand pumps " (fig. 29, *c*), &c. The well-known

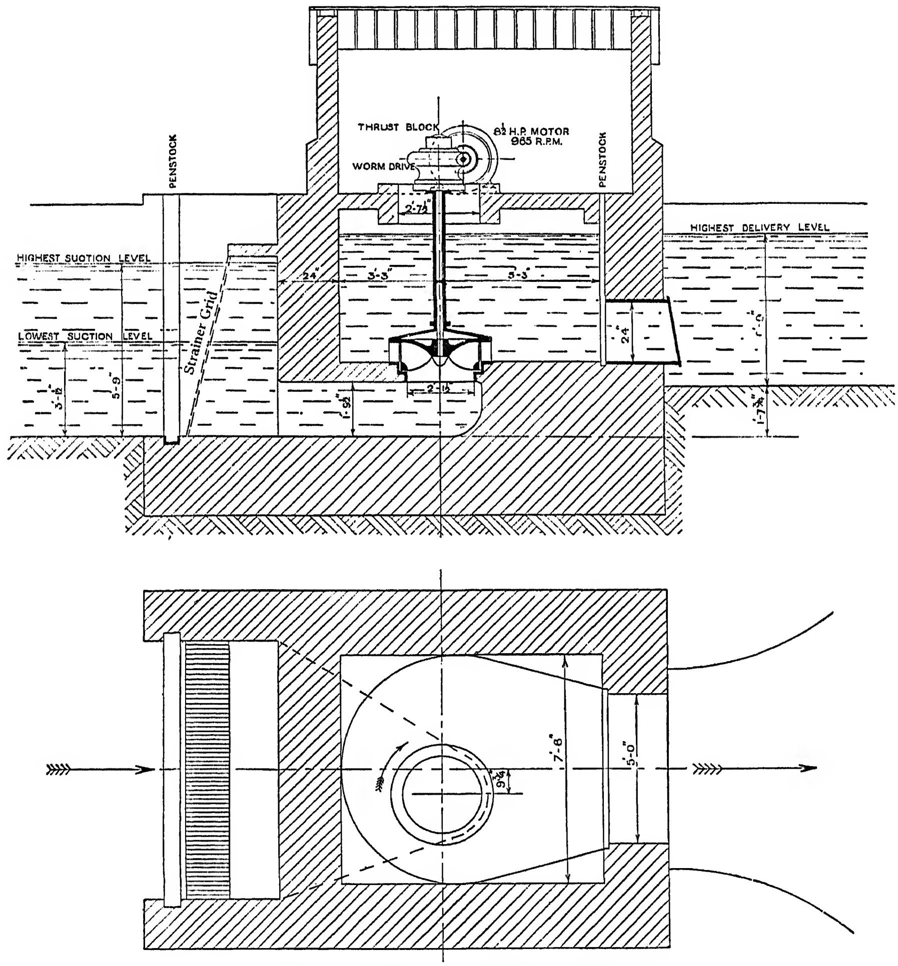


Fig. 31.—Open-type Pump Without Casing

Invincible pump made by Messrs. Gwynnes, Ltd., Hammersmith, and similar to fig. 28, *e*, and the Conqueror pump, made by Messrs. W. H. Allen, Son, & Co., Ltd., Bedford, and similar to fig. 28, *d*, are both low-lift pumps.

Pumps made by Messrs. Worthington, Simpson, Ltd., by the Pulsometer Co., Ltd., Reading, and by Mather & Platt, Ltd., Manchester, form very good examples of the double-cover pump with parallel bore casing as drawn

in fig. 28, *f*. The horizontally divided casing, fig. 28, *c*, is made by Messrs. A. S. Cameron Pump Works, New York, Messrs. Dayton-Dowd, Quincy, Illinois, and many other American firms; and quite recently by Messrs. Mather & Platt, Ltd., Manchester. For marine work and some land work, however, it has been common in this country for very many years.

Marine Circulating Pumps.—Centrifugal pumps for the circulation of sea-water through the condensers on board ship are usually known as marine pumps. Their general construction is similar to fig. 28, *d*, with the casing split horizontally or at an angle convenient for the suction and delivery branches. The method of running the impeller in the volute is adopted in order to reduce weight and over-all dimensions.

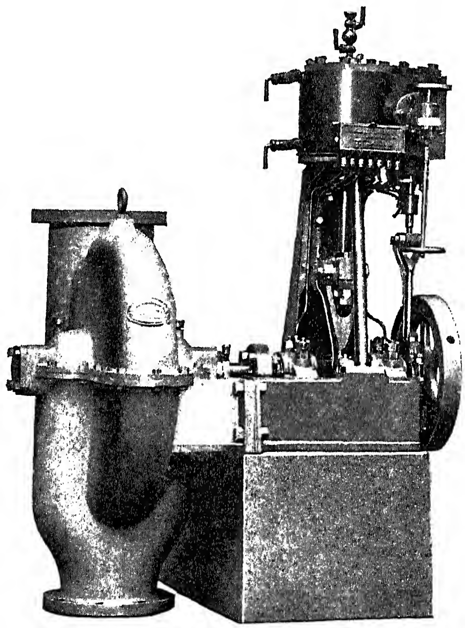


Fig. 32.—Small Marine Circulating Pump

Marine pumps are always below the water line, and therefore priming is automatic. The head they have to overcome varies between 18 and 35 ft., and is almost entirely friction head, the greater portion of which is due to the resistance in the condenser itself. The range of quantities dealt with by these marine pumps varies from a few hundred gallons per minute to upwards of 25,000 gall. per minute per pump.

The several features of the pumps and driving engines vary with the size of the unit. A type commonly fitted to cargo vessels is illustrated by fig. 32 from the practice of Messrs. W. H. Allen, Son, & Co., Ltd., Bedford; here the engine is of the single-cylinder open type, and the pump of the form already indicated. For larger units it is now very general practice to install enclosed forced-lubrication engines for driving purposes, as on account of their higher speed they enable much lighter sets to be built for any given output. High-pressure single-crank engines are used for small and intermediate sizes, and double-crank simple or compound engines for the larger sizes. An example of the latter is shown in the drawings, figs. 33 and 33 *a* of a marine circulating pump, manufactured by Messrs. Matthew Paul & Co., Ltd., Dumbarton. The unit was designed for an output of 18,000 gall. per minute against a gross total head of 35 ft. when running at 420 r.p.m., and consuming about 320 b.h.p.

A frequent arrangement in modern installations is for each pump to be supplied with two engines—one on each side—and each capable of driving the pump at normal full load, the other engine acting as a stand-by. A

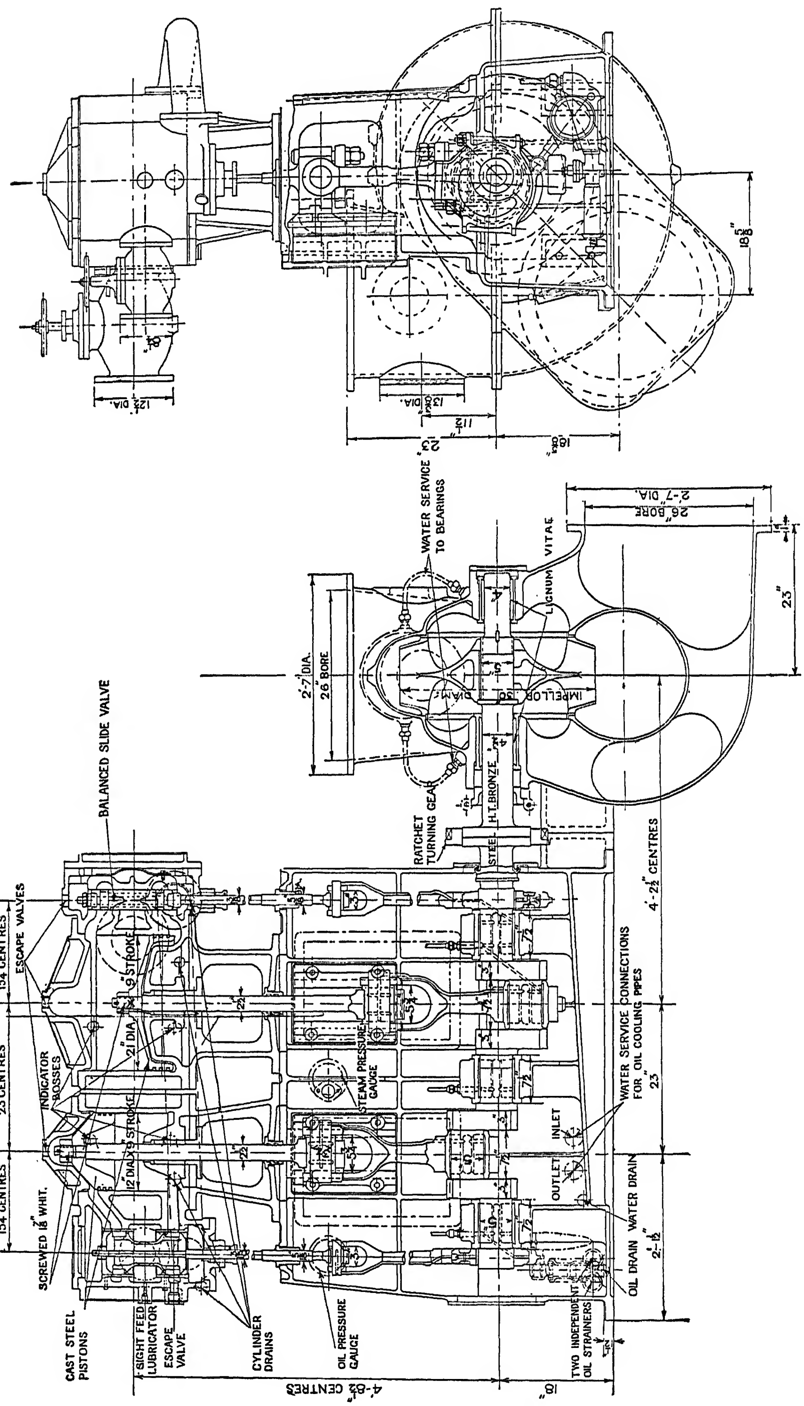


Fig. 33.—Compound Marine Pumping Engine

modification of this arrangement as made by Messrs. Matthew Paul consists of a pump with an engine on each side, exactly similar except for the cylinder diameter, each engine singly being capable of developing about two-thirds of the maximum power required, and the pair when linked together being capable of running as a compound engine and so effecting a considerable economy in steam consumption.

For ordinary mercantile work it is usual to fit cast-iron pump casings and gun-metal impellers, but for Admiralty work gun-metal pump casings are generally supplied.

Medium-lift Centrifugal Pumps.—Fig. 34 shows a double-suction

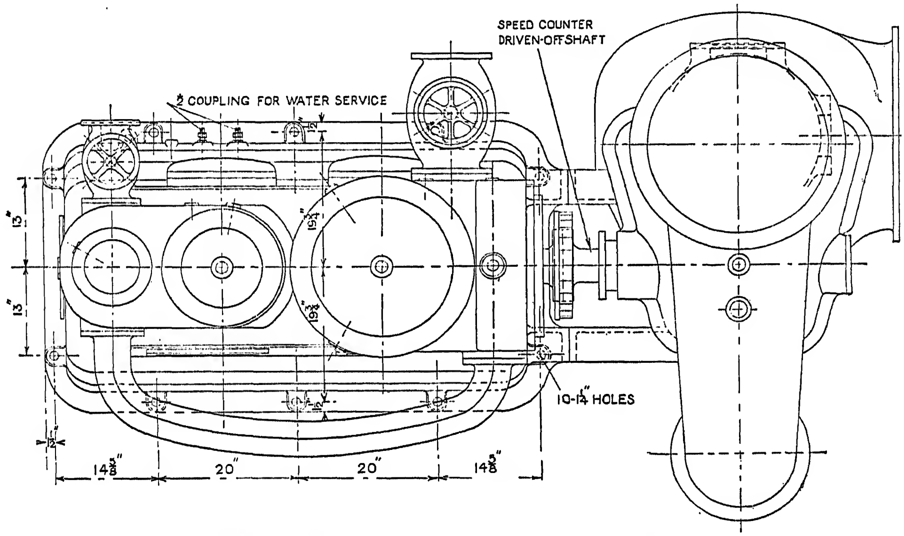


Fig. 33 *a*.—Plan of Compound Marine Pumping Engine

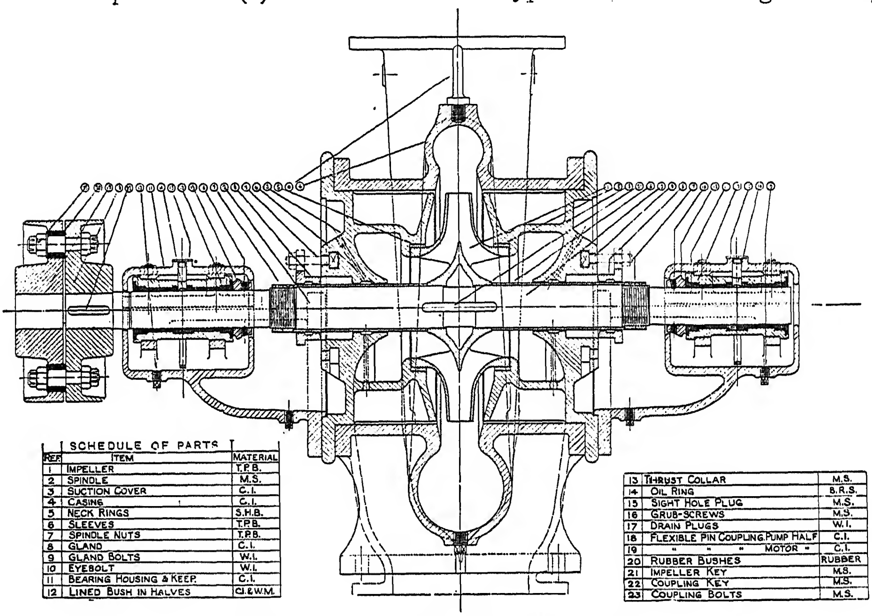
medium-lift pump as made by Messrs. Mather & Platt, Ltd., Manchester. These pumps work very efficiently for heads up to 150 ft. when the quantities are about 1000 imperial gallons per minute upwards; for smaller quantities it is not usual to supply this type for heads over 100 ft. The construction is clearly shown by the drawing.

Several manufacturers modify this design by inserting guide vanes between the impeller and the volute, so producing a single-stage turbine pump. In this form heads up to 300 ft. can be dealt with, and when the quantities of water are large enough 400 or 500 ft. head is economically reached.

High-lift or Turbine Pumps.—The term "turbine pump" is applied to guide-vane pumps irrespective of the head for which they are designed.

As regards the multi-stage designs these may be discussed briefly. The general type is shown in fig. 29, *a*. Here the casing is of the cylindrical form as further shown in fig. 35. Instead of this construction the outer shell may be split horizontally as seen in fig. 38, or it may be sectionalized into annular portions as shown in fig. 40. All these types have their advocates, and

while they may be used indifferently for many services, yet it sometimes happens that one of the types is more suitable than the others for a certain specified service. Presented with the bald fact of the several types, and asked to differentiate between their merits and demerits, it is difficult to do so without exhibiting some bias. It is perhaps enough to say that (1) the horizontally divided casing has an advantage in accessibility; its disadvantage is that internal stage leakage cannot be detected, and is exceedingly difficult to provide against. A further difficulty which occurs only in design is that when high pressures are dealt with the bolting of the horizontal joint presents a serious problem. (2) The sectionalized type has the advantage in easy



SCHEDULE OF PARTS

Ref.	Item	Material
1	Impeller	T.P.B.
2	Spindle	M.S.
3	Suction Cover	C.I.
4	Casing	C.I.
5	Neck Rings	S.H.B.
6	Sleeves	T.P.B.
7	Spindle Nuts	T.P.B.
8	Gland	C.I.
9	Gland Bolts	W.I.
10	Eyebolt	W.I.
11	Bearing Housing & Keep	C.I.
12	Lined Bush in Halves	C.I. & W.M.
13	Thrust Collar	M.S.
14	Oil Ring	B.R.S.
15	Sight Hole Plug	M.S.
16	Grub-screws	M.S.
17	Drain Plugs	W.I.
18	Flexible Pin Coupling. Pump Half	C.I.
19	" " " Motor "	C.I.
20	Rubber Bushes	Rubber
21	Impeller Key	M.S.
22	Coupling Key	M.S.
23	Coupling Bolts	M.S.

Fig. 34.—Medium-lift Pump built by Mather & Platt

manufacture, as a number of small light parts identical in detail, and suitable for repetition production are substituted for a single heavy part. Its disadvantage is that the reassembling of a dismantled pump requires greater care, as alignment is apt to become disturbed. (3) The cylindrical casing has the advantage of great rigidity and simplicity of dismantling, with the disadvantage that the cost is generally greater than the sectionalized type, and in certain circumstances, where a pump is not dismantled for a long period, there is a possibility of a little trouble in withdrawing the internal parts.

The single-sided impeller arranged in a simple series is now current practice on all multi-stage pumps for small and moderate quantities. Double-sided impellers are used, of course, when the pumps are sufficiently large to justify the extra complication.

In early forms of high-pressure turbine pumps axial thrust was a very serious difficulty; in modern designs the difficulty has quite disappeared, the "axial balancing device" is a practical success, and is now looked upon

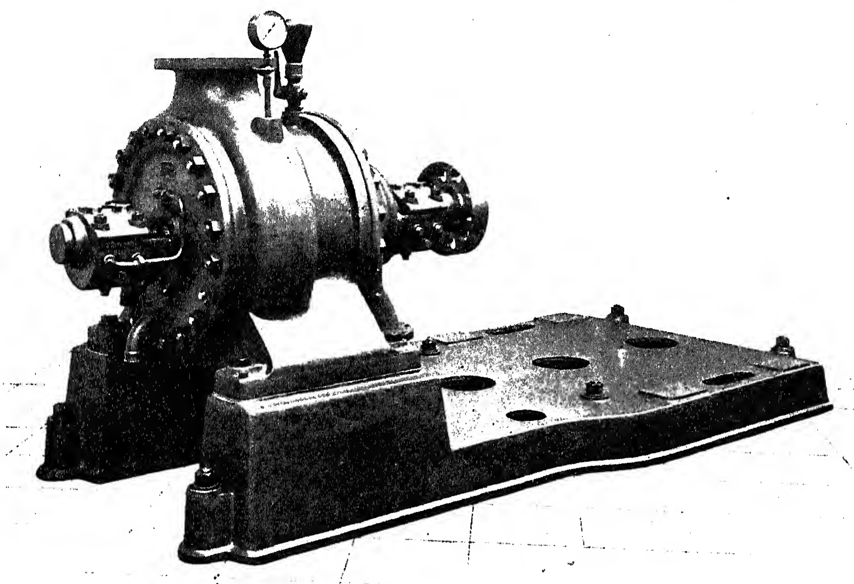


Fig. 35.—Multi-stage Turbine Pump built by Messrs. Escher. Wyss, & Co., Zurich
Duty: 1250 gall. per minute, 800 ft. head 1740 r.p.m., 76 per cent efficiency, 405 b.h.p

as a valuable asset in eliminating the high-pressure stuffing box. Such troubles as may now be associated with modern hydraulic balancers are identical with those that threaten every moving part of any high-speed machine.

With regard to the actual construction of the balancing element, a number of devices are described in a recent paper on "The Construction of Turbine Pumps" * by the authors, and as each form is practically a proprietary design, any particular type can hardly be singled out for description. The principle of design of all balancers is to provide a central circular aperture from the main pressure chamber of the pump, and to close this aperture by some form of rotating member arranged to open in the direction opposite to that of the axial thrust to be overcome. Automatic action is then secured by a proper control of the leaking water as it escapes from the pressure chamber through the rotating cover. A very simple, effective, and common form of balancer is shown in fig. 39.

Mechanical details such as bearings, spindles, impellers, &c., and the materials from which they should be made, are all discussed in the paper to which reference has just been made.

Turbine Pump Practice

One of the commonest and most important duties for which multi-stage turbine pumps are installed is that of draining mines. For this

* *Proceedings of the Institution of Mechanical Engineers*, May, 1917, and printed in various technical periodicals of the same date.

work they are suitable in every respect; they are light to install, occupy small space, and are suitable for direct coupling to electric motors. Fig. 35 shows a typical mine pump built by Messrs. Escher, Wyss, & Co., Zurich. The bedplate is made suitable for the driving motor, and transmission is by means of a balanced flexible coupling of the rubber pad and pin type. It is now general practice to drive from the suction end of the pump, as this permits of dismantling the whole of the interior without disturbing any pipe joints, as may be seen on reference to the illustration; also when many stages are necessary in order to overcome the required head, it is usual to divide the pump into two units, and place the electric motor between the pair.

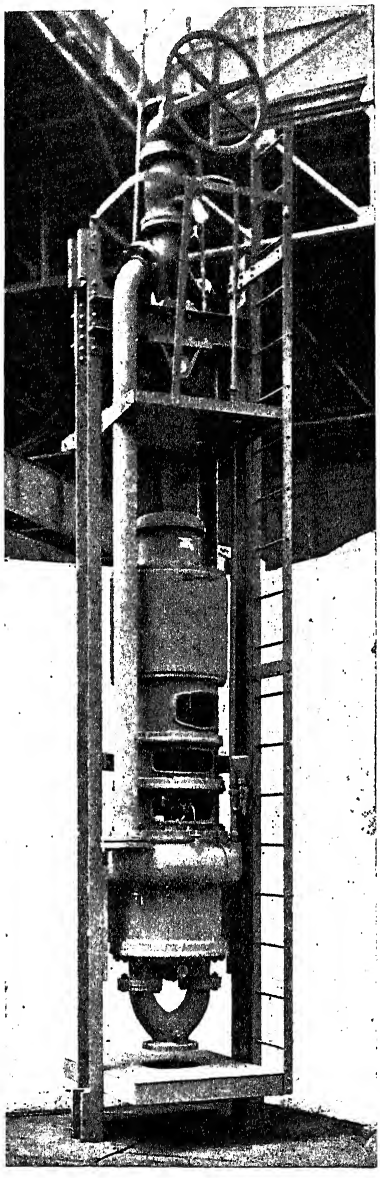


Fig. 36.—Sinking Pump built by Messrs. Escher, Wyss, & Co., Zurich

Duty: 265 gall. per minute, 460 ft. head, 1450 r.p.m.

Turbine pumps are particularly useful during shaft-sinking operations in mining. For sloping shafts the pump and motor are mounted on a low trolley, and for vertical shafts the unit is built into a steel frame provided with means for slinging. In both cases an exceedingly light pumping unit is obtained, and one which is easily lowered to keep pace with the water or with the rate of excavation. A sinking pump of the slinging type for a vertical shaft is shown in fig. 36. Generally speaking, sinking pumps are not built for higher heads than 1000 ft. or so. Limitations are imposed by the necessity for lightness, and by the fact that the long vertical delivery pipe has also to be slung, entirely or in part, by the same means as the pump. These handy pumping units, though primarily designed as portable machines, are often eventually bolted down for continuous duty when sinking operations are completed.

As boiler-feed pumps, the turbine or centrifugal pump is now well established. If the duty is to feed a single boiler, then the necessary amount of water to be pumped is too small to permit of the installation of a sufficiently large

pump to be reasonably practical. If, however, a battery of boilers is to be fed, and if, at the same time, either electric motor or steam turbine drive is permissible, all the elements of a highly satisfactory turbo feed-pump equipment are present.

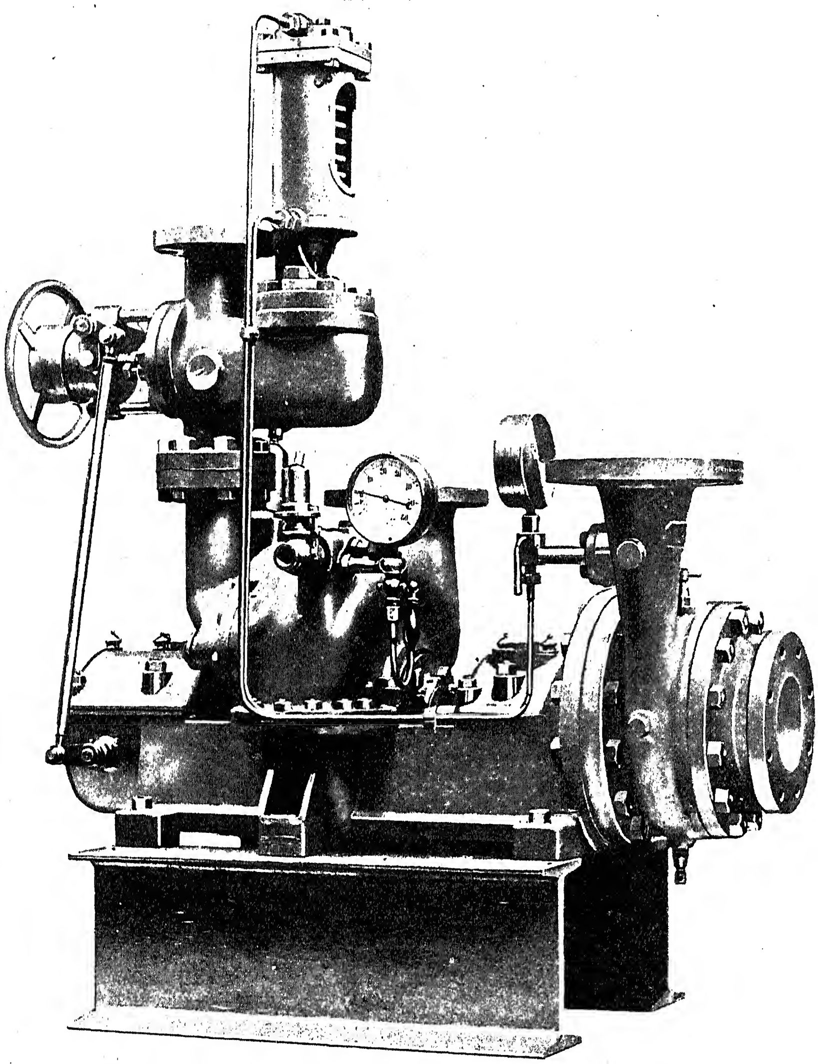


Fig. 37.—Boiler-feed Pump, Centrifugal Type—Steam turbine driven—built by Messrs. G. & J. Weir, Ltd., Glasgow

A single-stage form of centrifugal feed pump as built by Messrs. G. & J. Weir, Ltd., Glasgow, is illustrated in fig. 37. The object of this design is to produce the most economical combined set of steam turbine and pump suitable for boiler-feeding purposes. In determining the pump dimensions,

therefore, consideration has been given to the requirements of the steam turbine and vice versa. The turbine is of the impulse type with one pressure stage and several velocity stages, and the pump is of simple volute form—without guide vanes—but with special automatic axial balancer arranged to absorb hydraulically the whole of the end thrusts which may come on the rotor, either from the steam end or the water end. On account of the high pressure generated by the single quickly rotating impeller, special labyrinth rings are necessary at the running shoulders of the impeller to avoid what might otherwise be a serious leakage loss.

A duty for which multi-stage centrifugal pumps are particularly adapted is that of boosting up the pressures in town water mains during times of

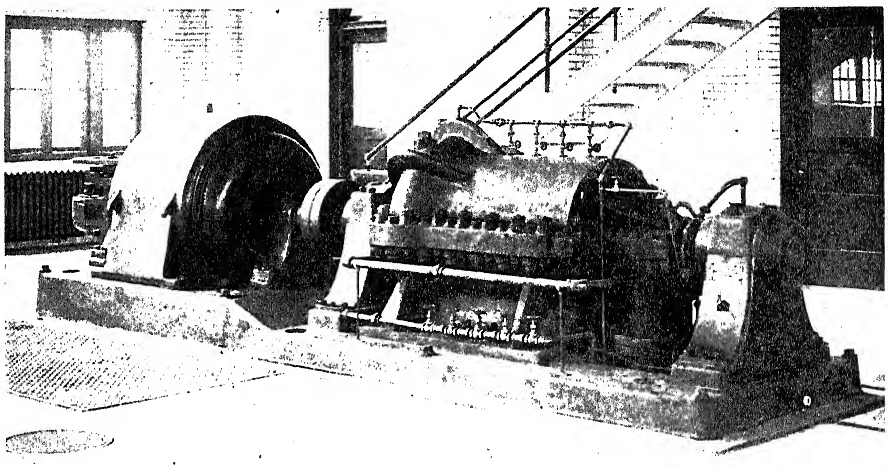


Fig. 38.—One of four 10-in. High-pressure Fire Service Centrifugal Pumping Units built by Allis-Chalmers Manufacturing Company for City of Cleveland

Rated capacity 2500 gall. per minute against 240 lb. pressure.

emergency, when water is being drawn for fire-extinguishing purposes. One of the high-pressure fire service pumps, built by Messrs. Allis-Chalmers Manufacturing Co., Milwaukee, for the city of Cleveland, is shown in fig. 38. Four of these pumps were installed with 10-in.-diameter branches, rated for 2100 imperial gallons per minute, against a delivery pressure of 240 lb. per square inch. The general construction of Messrs. Allis-Chalmers' pumps is shown in fig. 39, which is fully indexed to show the various parts. The axial balancing device here shown is one of the most generally used and simplest forms made, and when properly constructed is very reliable in action.

Bore-hole pumping, when the water is at great depths, is rather beyond the scope of the multi-stage turbine pump at its present stage of development. Practical reliability is of first importance in pumping plant, and up to the present it has not been found advisable to carry rapidly rotating vertical shafts to greater depths than about 100 ft. The difficulty is the great weight of the vertical shaft, and the risk of alignment troubles in the many guide

bearings, particularly as the shafts must run at least up to 1200 r.p.m. to give a useful effect. Messrs. Sulzer Bros., Winterthur, have quite a number of bore-hole pumps running very successfully at depths of about 100 ft. With bore-holes of about 20 in. diameter they can easily raise 330 imperial gallons per minute, and the whole plant, fitted with a vertical electric motor immediately over the bore-hole, forms a very compact and highly economical equipment.

Though strongly advocated over sixteen years ago, it is only within the last few years that the multi-stage pump has been confidently installed to

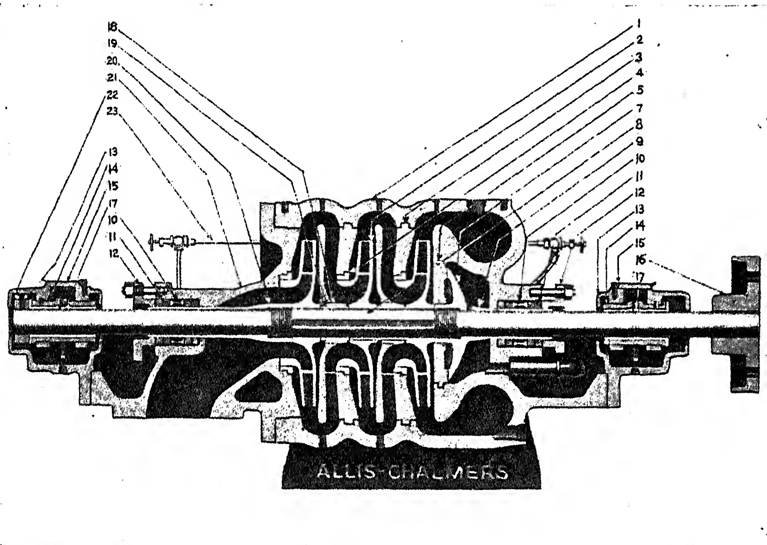


Fig. 39.—Section through Type "ST" Multi-stage Centrifugal Pump

1. Cast-iron casing. 2. Bronze enclosed runner. 3. Cast-iron return guide. 4. Bronze wearing ring. 5. Cast-iron thrust plate. 7. Balancing disc. 8. Open hearth steel shaft. 9. Bronze shaft sleeve. 10. Bronze water seal ring. 11. Packing. 12. Cast-iron glands. 13. Cast-iron bearing cap. 14. Cast-iron babbitted shell. 15. Cast-iron hinged lid. 16. Cast-iron flexible coupling. 17. Bronze oiling ring. 18. Bronze return guide bushing. 19. Bronze spacer sleeve. 20. Bronze shaft sleeve. 21. Cast-iron suction cover. 22. Cast-iron set collar. 23. Water seal piping.

supply hydraulic pressure systems. For this work they are particularly adapted, and in most cases an accumulator is quite superfluous, as the pump adjusts itself to the amount of water drawn from the system whilst continuing to run at a constant speed, and maintaining the pressure practically constant. The hydraulic pressures used by different consumers vary from about 400 to 1500 lb. per square inch, but 750 to 800 lb. is very general practice, and admirably suits a multi-stage turbine pump. For these duties an ordinary high-pressure multi-stage turbine pump, fig. 40, is generally employed without any special changes in design.

Positive Displacement Rotary Pumps.—Very many forms of positive displacement rotary pumps have been proposed, but in current practice only a few simple types are in general use, and the applications of these are

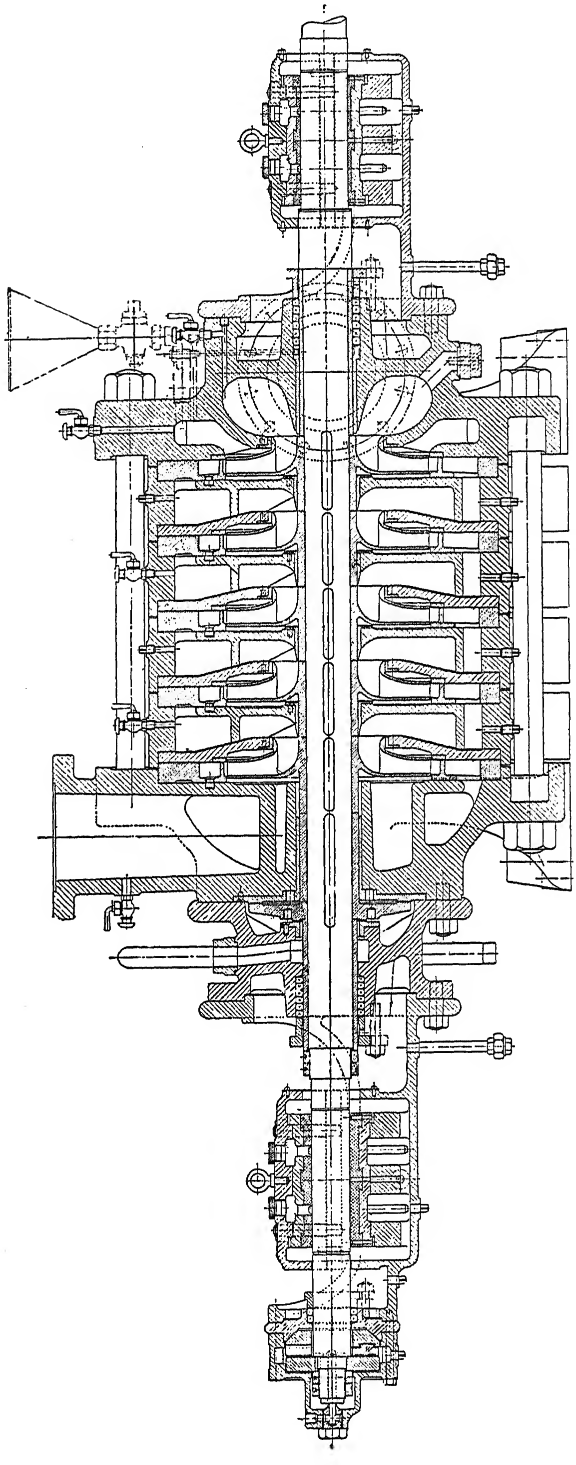


Fig. 40.—Sectional Arrangement of Hydraulic Pressure Pump built by Messrs. Mather & Platt, Ltd., Manchester

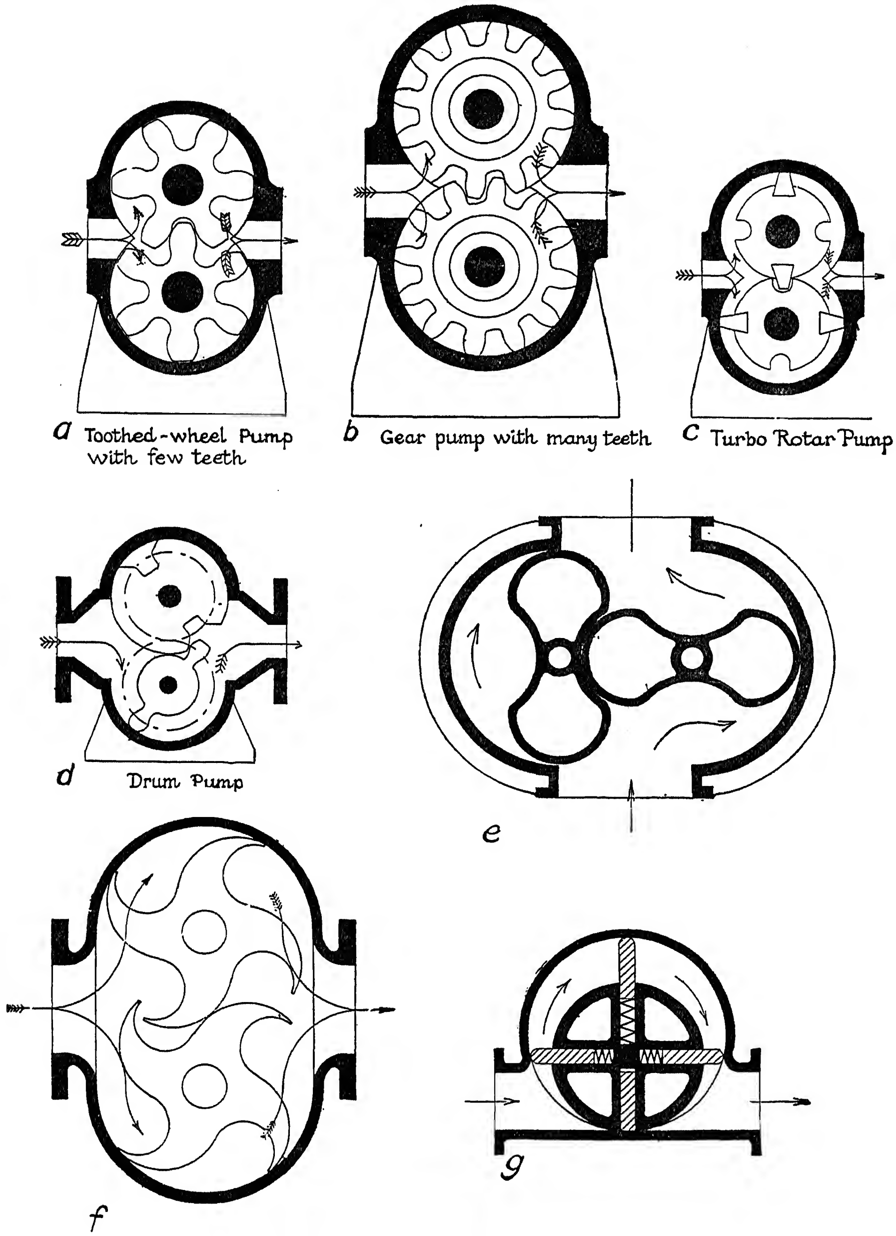


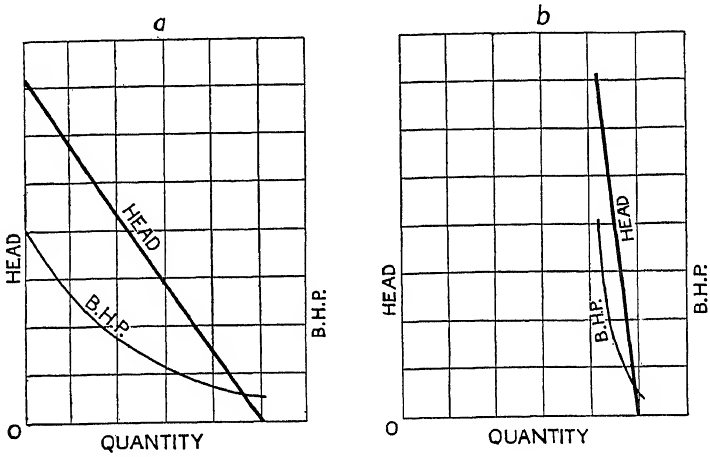
Fig. 41.—Principal Types of Rotary Pumps

restricted to certain limited duties. Owing to losses by hydraulic shock, rotary pumps in large sizes are not so successful when dealing with water as with air or gas, consequently it will be found that for water comparatively small sizes only are constructed on this principle, and in the exceptional cases where large units have been built, the pumping head has not exceeded

30 or 40 ft. On account of their comparatively slow speed, all of these small pumps deal equally well with water, oils, semi-fluids, acids, tar, &c

Rotary pumps may be divided into two classes:

1. Those consisting of rotating lobes, or meshed drums; and
2. Those with some form of revolving blade which either slides in its housing as it rotates or pushes some movable abutment out of its path each time it completes a cycle.



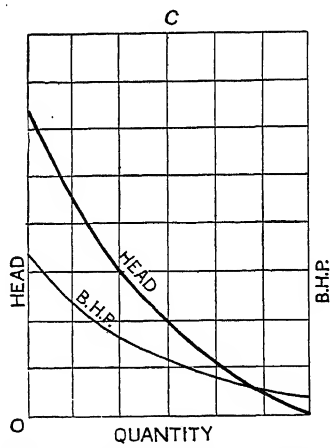


Fig. 42.—Characteristic Behaviour of Rotary Pumps at Constant Speed

(*a*) Gear pump with few teeth. (*b*) Gear pump with many teeth. (*c*) Pump with two teeth or two blades.

Most of the common forms in practical use are illustrated diagrammatically in fig. 41, *a* to *g*, of which *a* and *b* are self-driving, but *c*, *d*, *e*, and *f* require additional spur gearing between the spindles. There are forms with three meshing drums, but they are very little, if at all, used at present. Type *g* has one spindle only.

Behaviour of Rotary Pumps.—The performance of rotary pumps does not lend itself to scientific treatment, but the general characteristics of their behaviour are readily understood. Fig. 42, *a*, shows a typical performance curve, throughout its whole range of possible quantities and pressures, from a pump similar in design to fig. 41, *a*, and running at a fixed speed. When the pump is giving no delivery (or running with a closed delivery valve), the whole of the water displaced by the rotors is bypassed back as leakage, which occurs (1) between the teeth in mesh, (2) across the tips of the teeth

where they rub on the casing, and (3) across the ends of the teeth and the flat sides of the rotor. The pressure given by the pump at this condition depends therefore upon the relative proportion of the total leakage clearance to the volumetric displacement, and also upon the number of interruptions there are to leakage between the outlet and inlet of the pump. Thus, generally speaking, a rotor with many teeth will give a higher maximum pressure than one of similar displacement with few teeth, or a rotor that sweeps out a large volume in proportion to its leakage clearance will give a higher pressure than one of similar profile, but sweeping out a smaller volume.

If the pump is actually delivering against a pressure less than this no-delivery pressure, a portion of its displacement only is wasted in leakage, and the remainder is pumped. Thus both pressure and volume delivered depend entirely upon the accuracy of mechanical fit of the rotors with each other and with the casing, and—when the pump has been in use for some time—upon the condition of wear of the parts.

In larger sizes of pump the leakage clearances always have a smaller value in relation to the volume displaced, and such pumps therefore have an advantage in economical working.

The shape of the head-quantity characteristic differs very considerably with different forms of pump. With some types the shape becomes almost a vertical line, fig. 42, *b*, and it would be impossible to drive them with a closed delivery valve without bursting the pump casing. On other forms the curve approximates to the parabola that theoretical considerations would suggest, fig. 42, *c*.

The viscosity of the liquid being pumped has a very marked effect upon the maximum pressure, and the volumetric efficiency attained by any rotary pump. The ideal liquid is, of course, one which has good lubricating qualities, and also a comparatively high viscosity. For this reason oils are very suitably dealt with by the majority of rotary pumps.

Many manufacturers make toothed wheel or gear pumps * (fig. 41, *a* and *b*) in small sizes for pumping suds or oil and for small quantities of water to pressures not generally exceeding 100 ft. Usually the teeth are identical with straight spur gearing, but in some cases skewed or helical teeth are used, and in one pump of German manufacture the axes are set opposite to each other but slightly out of line, and bevel wheels are used. Gear pumps are not capable of running at a high speed or serious vibration is set up due to water-hammer caused by the entering teeth striking, and suddenly forcing out, the water contained in the spaces between the teeth of the opposite rotor.

Pumps of the type fig. 41, *c*, are made by Messrs. Pumps, Ltd., Birmingham, under the trade name of "Turbo Rotar" pumps. In the normal pattern the necessary gear wheels for rotating the drums are actually running in water, but for higher heads external gears are used. The vanes are of gun-metal, fitted into iron drums.

* This form of pump is known as "Servière's Rotary Pump", as its invention is usually attributed to Servière, a Frenchman born in 1593.

The form of pump shown in fig. 41, *d*, is made by Messrs. The "Drum" Engineering Co., Ltd., Bradford, in a wide variety of sizes and types.

A modified form of "Roots" revolver (fig. 41, *e*) is made by Messrs. Samuelson & Co., Ltd., Banbury, in their "Acme" rotary pumps. These are designed for a range of heads up to 250 ft., and for capacities to 50,000 imperial gallons per hour. The illustration, fig. 43, shows one of these pumps designed for delivering 42,000 gall. per hour to a head of 140 ft., and for use in a coal-mine.

Rotary pumps are carried to large sizes in the Roots type, where there are two rotors each consisting of a two-lobed cam. The Connersville Blower

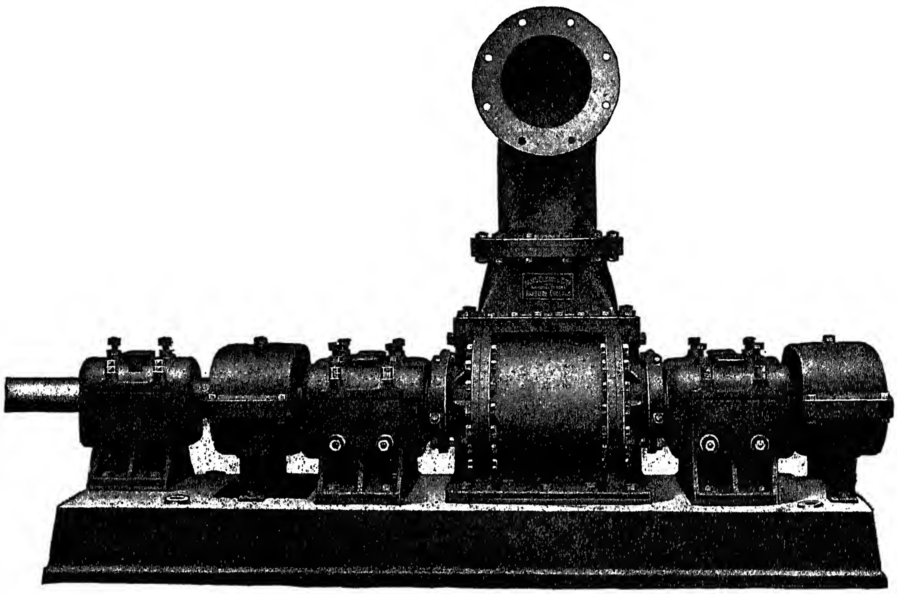


Fig. 43.—" Acme " Rotary Pump

Company, Connersville, United States, have built units to deliver $1\frac{3}{4}$ million gallons an hour against heads of 30 to 35 ft., and a number of such plants direct-coupled to Corliss engines are in operation for irrigation purposes in Louisiana and Texas. A pump with revolvers 58 in. diameter and 52 in. wide, running at 55 r.p.m., gave a volumetric efficiency of 96 per cent, and a mechanical efficiency in the neighbourhood of 85 per cent.

The sliding-blade type of pump, fig. 41, *g*, is the original form* of a distinct class of rotaries, in all of which one rotor is set eccentrically relative to the other. In the pump drawn the blades, which may be one, two, three, or four in number, constitute one rotor, and the eccentric drum is the other rotor. This construction is still used for pumping water, though its manufacture is chiefly carried out on the Continent.

Examples of rotary pumps with curved arms similar to fig. 41, *f*, are manu-

* Often known as "Ramelli's Rotary Pump". Agostino Ramelli (1530–90) gives an illustration of this pump in his book published in 1588.

factured by Messrs. The Goulds Manufacturing Co., Ltd., Seneca Falls, New York, United States, who also make the gear form of pump, fig. 41, *b*, for smaller sizes.

The rotary gear pumps so far referred to are of the ordinary spur gear design, and it is interesting to consider what would be the result of seeking to utilize "internal gears" for pumping. The attempt has been made, and is now marketed as the "Feuerheerd" rotary pump by Messrs. Stothert & Pitt, Ltd., Bath. This device is simple and particularly sweet in its action. Water is admitted to the rotors through the roots of the teeth of the outer geared rotor, therefore the water-hammer, already referred to, which occurs at high speeds with the ordinary gear pump, is altogether avoided.

A pump which, in its hydraulic action, is practically similar to the one just described, though very different mechanically, is the Rotoplunge pump, made by Messrs. The New Rotoplunge Pump Company, Westminster. Mechanically, this pump is a combination of the rotary and reciprocating types, as its name implies.

In common with most rotary pumps, the Rotoplunge is particularly well adapted for dealing with heavy oils, though it is also very successful with water. When pumping oil, heads up to 180 ft. can be overcome, but with water it is preferred not to exceed 100 ft. On the suction side a vacuum as high as 28 in. of mercury can be obtained.

FLUID IMPELLENT PUMPS

Pulsating Pumps. — Amongst modern pumps very many forms of automatic pulsator or pulsometer pumps are to be found, some operated by steam, others by compressed air, and in the latest newcomer by the pressure of combustible gas.

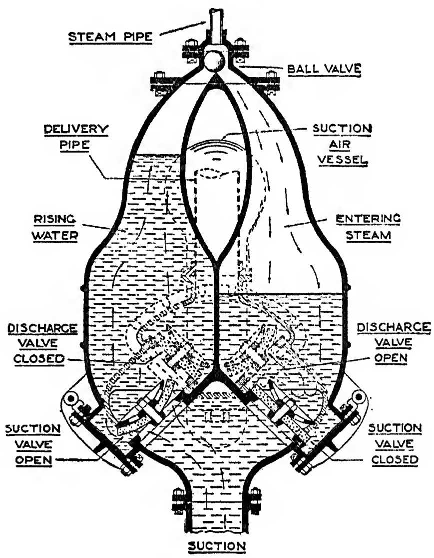


Fig. 44.—Section of Pulsometer Pump

Pulsometer Pumps. — As the general behaviour of all pulsometers is practically identical, it is only necessary to explain one of them in detail. Fig. 44 shows a section of a pulsometer steam pump as made by Messrs. The Pulsometer Engineering Co., Ltd., Reading. The apparatus consists of a double-chamber body, each chamber being provided with a suction and delivery valve, and both being controlled by a ball steam valve. For easy starting it is usual to charge the pump by filling both chambers with water, though many pulsating pumps

are self-filling for low-suction lifts. When the steam valve is opened, steam will then pass to one or the other of the chambers, expelling the water by virtue of the steam pressure. As the steam fills the chamber it encounters the increasingly large cold surface walls of the chamber, and also a larger area of water surface, both of which cause considerable condensation. Then as the water is further depressed till it reaches the level of the outlet valve, some steam escapes through the delivery valve, and the water surface is ruffled and agitated, so that condensation of the steam progresses so rapidly that a partial vacuum is formed, thereby causing the ball valve in the steam head to clap over and shut off the steam. Condensation then proceeds, and water is drawn into the pump chamber as the vacuum is formed. Meantime the steam has been diverted to the opposite pump chamber, and is expelling the water there, and so the action is continuous. Snifting valves are often provided at the top of the pump chamber to suck in a little air on the suction stroke, and so control the pulsations while at the same time acting as a buffer between the steam and the water, and thereby reducing waste condensation.

The improvements which have taken place in pulsator pumps relate to (1) the introduction of water jets, or rose sprays, which come into operation automatically as the outgoing water is depressed to a predetermined level, and thereby positively initiate the rapid condensation which finishes the delivery stroke, and causes the change-over of the steam valve. (2) The automatic control of the oscillating steam valve so as to produce a certain degree of expansive working, and also cut off the steam supply during the time that condensation is in rapid progress. This is done either by means of a sensitive auxiliary valve controlled by the fluctuation of steam pressure in the pump chamber as soon as condensation begins, or by fitting a shuttle valve in the steam head (instead of a simple clapper), the two ends of which are in communication with the lower ends of the pump barrels, the effect of which is to push the valve over immediately a drop of pressure begins towards the termination of the expulsion stroke. (3) The application of pulsometers to higher pumping heads by the introduction of the above refinements, combined with the employment of higher steam pressure and smaller pump chambers.

In practice, pulsometers are capable of dealing with suction heights as great as 25 ft. when fitted with short suction pipes and dealing with cold water, though it is safer, and a better output is obtained, with suction lifts not exceeding 10 to 14 ft. The maximum possible total height pumped depends upon the steam pressure, but about 100 ft. is the usual working maximum and 150 to 170 ft. for specially designed pumps with steam pressures of about 100 lb. per square inch. For higher heads the lift must be made by using several pulsometers pumping in stages.

There are two well-known empirical rules for determining the minimum steam pressure for any given lift. One is to add from 10 to 30 lb. per square inch to the maximum pressure height measured in pounds—10 lb., of course, for low pressures (say to 20 ft.), 20 lb. for medium pressures (say to 50 or 60

ft.), and 30 lb. for higher pressures. The other is to allow about 3 lb. steam pressure for every 4 ft. head on small pumps, and 5 lb. steam pressure for every 8 ft. head on larger pumps.

Roughly speaking, the steam consumption is proportional to the quantity of water delivered, and not sensibly dependent upon the head. Ordinary pulsometers will give about 6500 to 9500 ft.-lb. in water lifted per pound of steam, and improved pulsometers with automatically cut off expansion valves up to 14,000 ft.-lb. in water lifted.

The number of pulsations per minute, and consequently the quantity of water delivered, vary in inverse proportion to the head, and are greater the greater the steam pressure is in excess of the head. Experiments have shown a variation of from 15 to 50 pulsations per minute by control of the steam

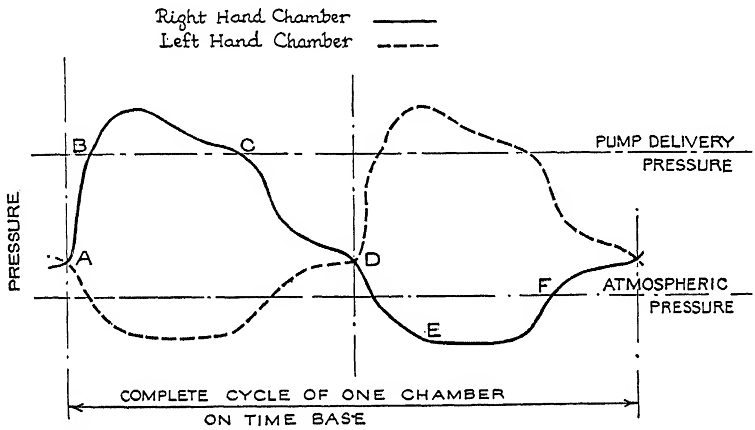


Fig. 45.—Pulsometer Steam Pump Chamber Pressure Diagram

pressure, with an average of about 40 pulsations per minute for the double chamber pump on 30 ft. head, the displacement being between 85 and 90 per cent of the chamber volume; on increasing the head to 70 ft. the volume discharged would diminish by about 20 per cent.

The thermal efficiency of a pulsometer is exceedingly low on account of the large quantity of heat given up to and carried away by the water, and also lost through the walls of the pump chambers. The actual amount of heating of the water pumped, however, is insignificant on account of the considerable volume passing; general experience shows the increase of temperature to be approximately 1° F. for every 6 to 8 ft. lifted.

The form of indicator diagram obtained from an ordinary pulsometer pump is shown in fig. 45. Such a diagram would be obtained on a drum driven by clockwork, with the pencil controlled by the pressure near the bottom of the pump chamber. At A steam enters the working chamber which is fully charged with water, and the pressure rises until point B is reached. Expulsion of water then commences until the point C is reached, at which condensation is beginning. Steam is not cut off, however, until the point D is reached, when, of course, the pressure is switched over into

the next chamber. The steam admitted between the points C and D is therefore all wasted, and with an automatically controlled steam valve and jet condensation as previously mentioned, this portion of the diagram C to D would be an almost vertical line, and obviously there would be a great economy of steam. From D to E a vacuum is formed owing to the completion of condensation, and the pump chamber is charged. At F the suction valve closes and the weight of the water is borne by the suction valves. The cycle is then completed, and the barrel is ready to come into operation as soon as the steam valve is clapped over by the action in the neighbouring chamber.

Modifications in the mechanical design of pulsometers are controlled by the same considerations as in other pumping machinery.

Rubber pump valves are suitable for most light cold liquids, and particularly water. If, however, the liquid is hot or thick (as for instance with

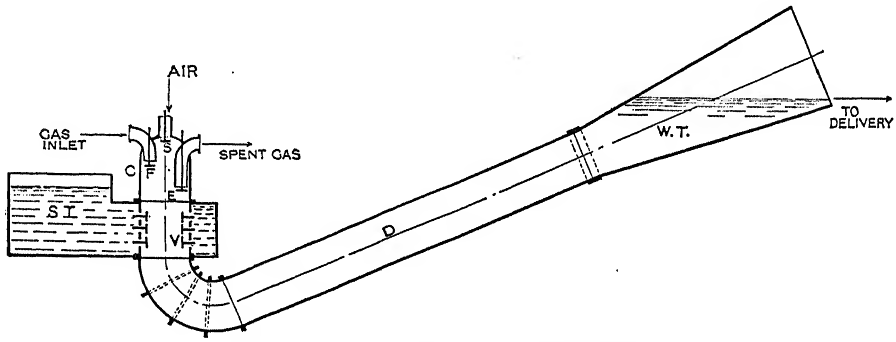


Fig. 46.—Diagrammatic Section of Humphrey Pump

tars, syrups, fats, &c.), metallic valves are preferable, and in many such cases ball valves are fitted.

For contracting work, unwatering pits and mines, &c., the great simplicity and convenience of the pulsometer is an advantage far outweighing its extravagance in steam, and in fact in more permanent duties there are many requirements which it can satisfactorily fulfil. For many duties the pump can be simply slung on a chain, and if permanently fixed on site quite light foundations supply all that is necessary.

The Humphrey Gas Pump.—Attempts to combine the simplicity of the pulsometer with the economy of the internal-combustion engine date back to 1868, but it was not until 1906–9 that the problem was successfully achieved. The realization of the object was due to Mr. H. A. Humphrey, and was brought about by the introduction of a swinging body of water, the periodic movement of which performed the cycle of operations necessary at the combustion end of the pump.

A diagrammatic section of the pistonless form of four-cycle Humphrey pump is shown in fig. 46. Constructionally the device consists of a suction tank ST communicating through spring-loaded suction valves V with the short arm of a U-pipe D (sometimes called the play pipe). The U-pipe terminates at one end in a combustion head C, fitted with exhaust valve E,

gas valve F, and scavenging valve S, and at the other end in a water tower WT communicating with the delivery water.

Bearing in mind the ordinary cycle of a four-stroke internal-combustion engine, the operation of the pump is easily followed.

Suppose a mixture of gas and air to be compressed into the top of the combustion chamber by the pressure of water in the long arm of the U-pipe, and then electrically fired. The explosion pressure sets the whole column of water in motion, and the continued pressure of the expanding gases adds momentum to the water. This large body of water cannot come to rest suddenly; its inertia carries it outwards towards the water tower until the pressure in the combustion chamber falls below that of the atmosphere. The exhaust valve E then opens by its own weight, the scavenging valve S is opened by the atmospheric pressure outside, and water is sucked in through the valves V from the supply tank. A light non-return valve fitted to the exhaust valve prevents previously expelled exhaust products from returning to the pump chamber. Its momentum being exhausted, the water flows back along the play pipe D, closing the scavenging valve, and driving the burnt gases out of the combustion chamber until the exhaust valve E is shut by the direct impact of the rising water. The trapped air and spent gases are then compressed into the combustion head to a pressure which, due to the work done in bringing the column of water to rest, is considerably in excess of the pumping head. Accordingly the water column makes a rebound outwards, again lowering the combustion chamber pressure below that of the atmosphere. This time the exhaust valve and scavenging valve are prevented from opening by a pressure-operated interlocking gear which simultaneously releases the inlet valve F, so that this opens and a charge of gas and air is admitted. The water column, having exhausted its momentum, returns inwards, compressing the new charge and firing it automatically by a pressure-operated ignition apparatus, thus starting a fresh series of operations.

This working cycle is shown clearly on the time-base diagram, fig. 47. The charge is compressed from A to B and fired at B, whence the pressure rises to C. Whilst the water column makes its long outward movement the gases expand from C to D where the exhaust valve opens, scavenging begins, and water enters through the suction valve. At E the water column reverses and exhaust continues to F, where cushioning begins and finishes at G when the water column comes to rest. The short rebound stroke of the water column then commences, and the cushion expands to H; here the new charge is admitted until J is reached, at which point the water column again reverses and compression begins ready for the next cycle. The corresponding thermodynamic cycle is given at the right-hand side of the time-base diagram, and corresponding points are similarly lettered.

The rate of working of a Humphrey pump depends upon the weight of water in the play pipe, and this is determined during the design, by varying the length of the pipe. The longer the pipe the greater the weight of the water column, and for a given volume of mixture in the charge the less will

be the velocity of the water column and therefore the slower the pump will work. The water velocity is inversely proportional to the square root of the pipe length; too short a pipe gives too high velocities and excessive friction losses, too long a pipe unnecessarily reduces the possible pump output.

Regulation of a pump's capacity, when installed, is obtained by varying the level of the water in the suction tank, thus causing it to work with a bigger or smaller charge of combustible mixture. The lower the suction level, within limits, the greater is the charge volume and output of the pump.

The outstanding feature of the Humphrey pump is its remarkably high efficiency, and the factors which contribute to this result are:

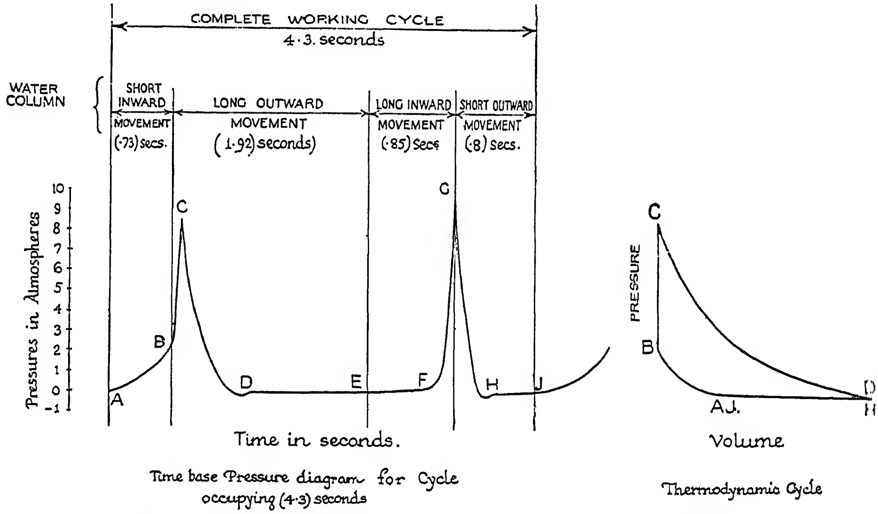


Fig. 47.—Humphrey Pump-pressure Diagram and Thermodynamic Cycle

1. The charge is expanded down to atmosphere, thus utilizing the "toe of the diagram" instead of exhausting at an absolute pressure of 45 lb. per square inch or so, as is done in the ordinary gas-engine.

2. The combustible charge is considerably stratified—that is, rich and rapid burning at the upper part of the combustion chamber, hence reducing the heat losses.

3. The explosion and expansion pressures act directly on the water pumped without the loss of power due to intermediaries, such as piston, connecting rod, bearings, gear, &c., which occur with other systems.

Tests carried out by Dr. W. Cawthorne Unwin, F.R.S., during September, 1909, on a 16-P.H.P. Humphrey pump showed a consumption of 1·06 lb. of anthracite coal per P.H.P. hour. More recently the large pumps installed by the London Water Board at Chingford, and of approximately 300 P.H.P., showed a consumption of about ·9 lb. of coal per P.H.P. hour. These results constitute a record in pumping economy, but it is noteworthy that considerable improvements on these figures are confidently anticipated as a result of further developments.

The Chingford pumps just referred to are the largest examples of Humphrey pumps yet installed, and illustrate the class of work for which the pump is eminently suitable. This installation consists of four units with 7-ft.-diameter cylinders each designed for 40 million gallons a day, and one unit with 5-ft. cylinders for 20 million gallons a day. One of the larger units is shown in elevation in fig. 48, from which the construction is seen to be as already described, but with additional accessories. The concrete air duct shown is for the supply of scavenging air, and provision is made for maintaining this duct at a pressure from $\frac{3}{4}$ lb. to $1\frac{1}{4}$ lb. per square inch to enable the pump to deal with considerable overload should this ever be necessary. Multiple valves are necessarily fitted for exhaust, inlet, scavenging, &c., and belt passages surround the combustion head communicating with these valves. The exhaust duct leads straight to the atmosphere, and as the burnt gases are exhausted at atmospheric pressure the pump works practically without noise. The water valves in the suction box are of the flap type with torsion springs fitted on the hinge pins, both valves and seats being of phosphor bronze.

The fuel to operate the pumps is supplied from four Dowson gas producers, three of which are rated to convert 370 lb. of anthracite per hour and the other one 138 lb. per hour.

The operation of the Humphrey pump is not confined to the four-cycle movement, and it is on the lines of the two-stroke cycle that Messrs. William Beardmore & Co., who have taken over the pump, are now (1922) energetically developing the invention. An important advantage of the two-stroke cycle is that higher compression pressures are possible, and therefore a reduction of fuel consumption will result, a figure of 0·55 lb. of coal per P.H.P. hour being within the range of possibility.

The pump is being adapted for oil fuel, also for heads as high as 100 to 150 ft., and is also capable of dealing with an actual lift on the suction side. Other attractive proposals are the utilization of the pump for air compression and for generation of power by passing the water pumped through water turbines.

Air-lift Pumps.—It is believed that certain bubbling springs and overflowing oil-bores are natural air-lift pumps in which air or other gas becomes entrained under pressure, and by virtue of its expansion and the free outlet the gas raises the overflowing liquid. In 1797 a German mining engineer, Karl E. Löscher, made some investigations into the use of compressed air for raising liquids, and in 1846 the method was successfully employed in raising petroleum from bore-holes in Pennsylvania. Subsequent pioneers of the air-lift system were, A. Brear, 1865; J. P. Frizell, 1880; and Dr. J. Pohlé, 1886. There was considerable diversity in the apparatus used by the different workers, but the underlying principle was the same in all cases, and the results differed only in the degree of their efficiency. A rising main was deeply immersed in the liquid to be raised, and compressed air then admitted to the lower end of the pipe. The air expanded as it rose in the rising main, with the result that the average density of the column of

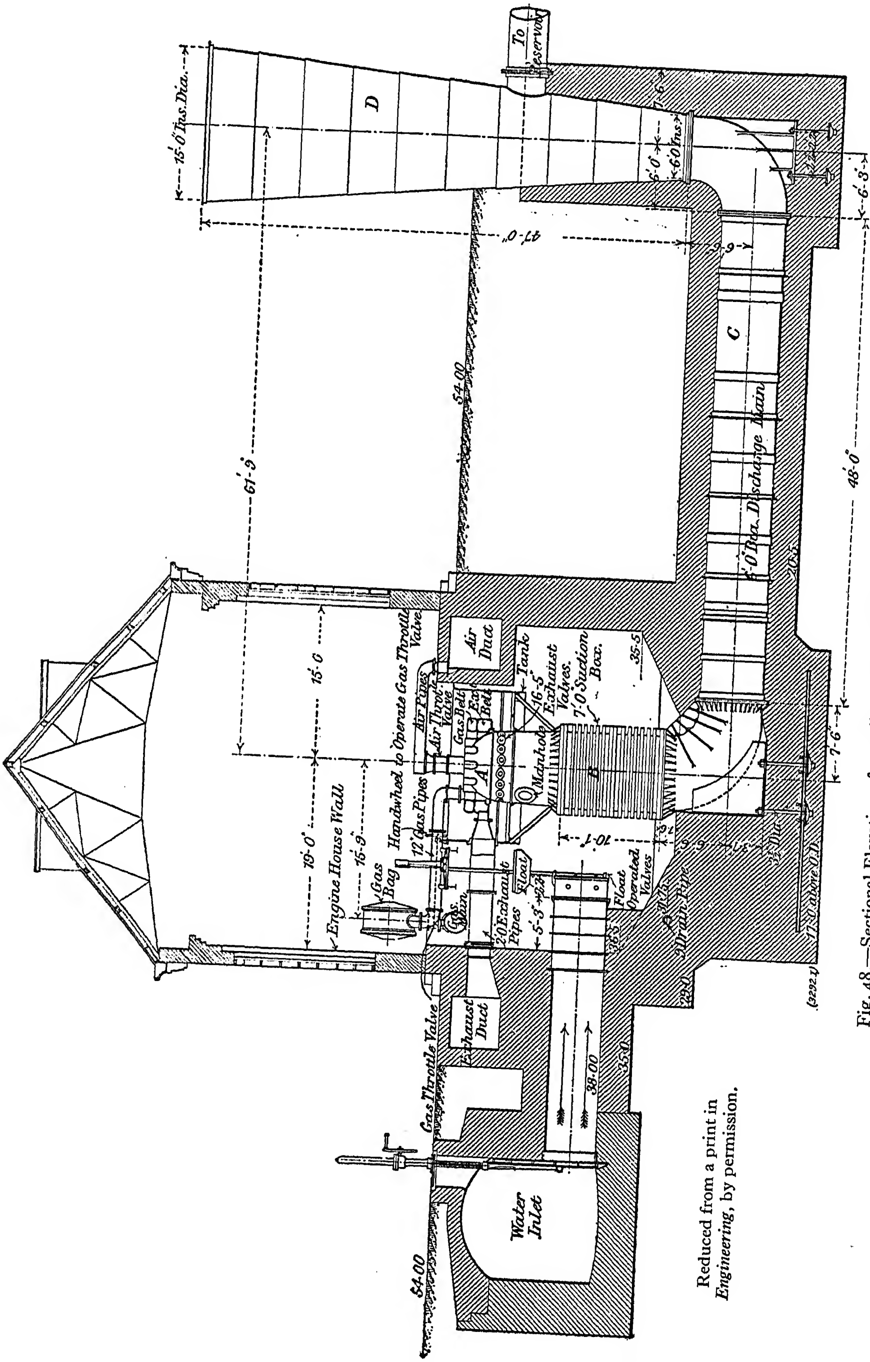


Fig. 48.—Sectional Elevation of 40-million-gallon Pump Installation at Chingford

Reduced from a print in *Engineering*, by permission.

aerated water was diminished, and its weight no longer balanced the weight of the surrounding water at the foot of the rising main. Consequently flow took place into the rising main, and the water level in the main was elevated. By correctly proportioning the depth of submergence, and the volume of air introduced, the aerated water was raised to any desired height. Considering the conditions at the foot of the rising main, it is obvious that the pressure due to the weight of mixture in the rising main must be practically equal to the pressure of the water surrounding the foot-piece, consequently, other things being equal, the greater the height to be lifted, the greater must be the depth of immersion of the rising main. The great depth of immersion necessary for high lifts sometimes proves a disadvantage, but in most of the useful applications of the air-lift pumps this factor is of little consequence. A valuable asset is the absence of mechanical parts in contact with the water, the compressor may be lodged in a convenient building and connected by a pipe to the bore-hole, and the whole process put in motion by simply opening a tap. The device is not affected by sand, and is therefore useful for clearing a bore-hole of sand, or in cases where sand cannot be excluded. There are no suction difficulties, corrosive liquid is easily handled, and with hot liquids the efficiency of the apparatus is very much improved on account of the greater expansion of the air. The limitations of the air-lift pump are that it cannot discharge into a long length of pipe, and it cannot empty a tank. Other disadvantageous factors are that the presence of air with the water promotes the corrosion and destruction of the rising main, and sometimes causes deposits of salts; also, though the aeration of the water and the sparkling appearance produced is sometimes considered advantageous, the large quantity of air passed through the water sometimes increases its bacterial content.

Fig. 49 shows a diagrammatic view of an air-lift pump and several varieties of foot-piece. There are a number of ways of piping the well—the air pipe may be quite separate from the rising main, it may be arranged down the centre of the main, or it may be disposed concentrically around the rising main. The first method is the cheapest and is common practice; the second method has the advantage that the air is cooled by the rising water, and also the method permits of using the largest possible rising main for a given bore-hole diameter, but an addition is made to the frictional resistance of the main; while the third method is discounted by its greater cost.

Experience seems to indicate that there is not much difference in the behaviour of different varieties of foot-piece, provided that the air is reasonably broken up on entering and if materials are used which are not subject to rapid corrosion. Attempts to produce an injector effect at the mouth-piece have shown a doubtful advantage.

As regards the design of air-lift pumps there is a lack of a simple comprehensive theory, and differences of opinion exist amongst authorities. Practical data derived from actual experience is therefore the most valuable guide in determining proportions. In this connection reference may be

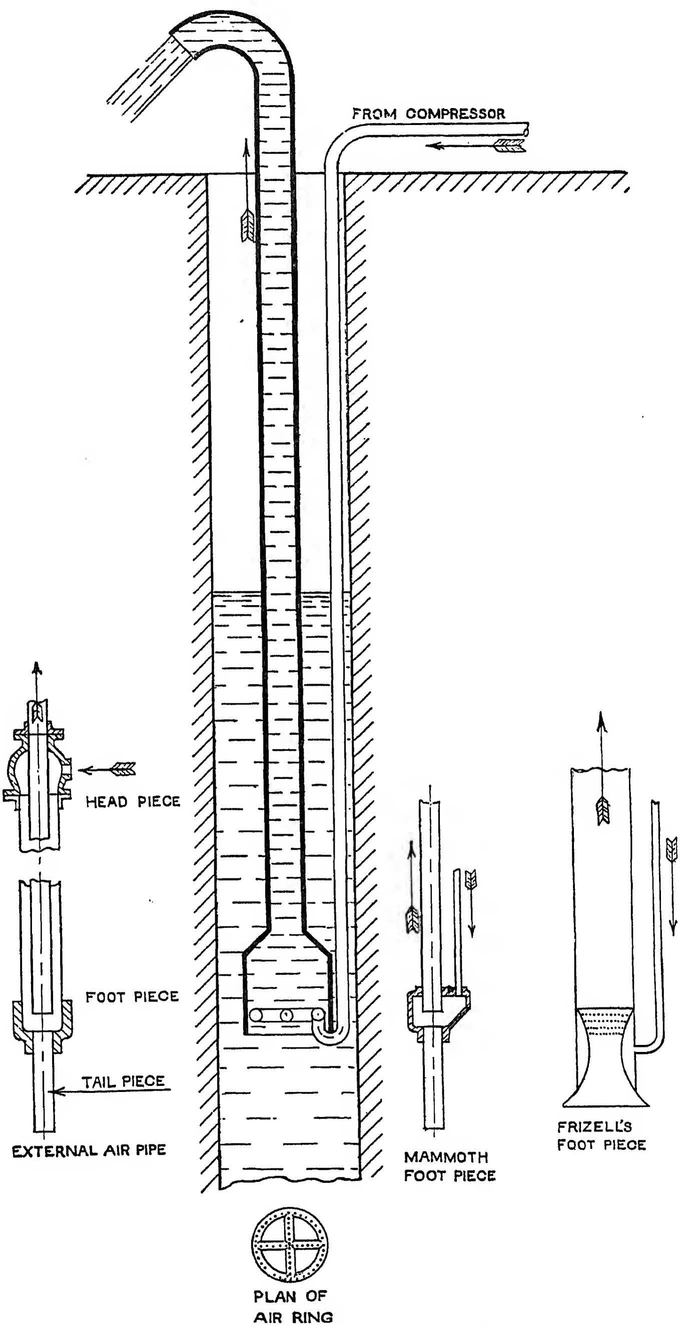


Fig. 49.—Air-lift Pump and Foot-pieces

made to a valuable paper * by Mr. A. W. Purchas, in which useful data is given, and the following table has been calculated from his recommendations.

Lift.	Ratio $\frac{\text{Submersion}}{\text{Lift}}$.	Submersion.	Approx. Ratio of Volumes = $\frac{\text{Atmospheric Air}}{\text{Water Raised}}$.
Feet.		Feet.	
25	4·06	100	·827
50	3·00	150	1·005
75	2·33	175	1·168
100	2·00	200	1·32
150	1·70	255	1·61
200	1·38	275	1·875
250	1·22	305	2·13
300	1·00	300	2·36

The depths of submersion here given represent a fair average of current practice; the ratio of air to water represents the volume of air necessary to raise unit volume of water against the particular lift. The latter values are calculated by a formula given by Mr. Purchas, but assuming that atmospheric temperature and the temperature of the water at the foot-piece are equal, the quantities are the net quantities required at the foot-piece, and do not allow for leakages, slip and losses in the compressor, &c. The figures are less than commonly supposed in practice, and if air volumes are calculated on the displacement of the compressor, quantities rather more than double those given will generally be necessary. The experimental results of Mr. Purchas and also of Dr. Jn. S. Owens (British Association Meeting, September, 1921) show that air volumes of from 2 to 2½ times the figures in the calculated table were actually employed.

As a result of his experiments Mr. Purchas arrives at the conclusion that high efficiency synchronizes with a low velocity of the rising mixture through the foot-piece, or else a low ratio by volume of air to water, or both simultaneously. This suggests as large a rising main as possible, and other experiments carried out at the Wisconsin University † indicate that a further important economy is secured by expanding the main towards the top. Further data on this point is desirable, as too large a rising main permits excessive slip of the water between the rising air bubbles; obviously, however, the bubbles have expanded considerably towards the top, and a greater sectional area of pipe is permissible there.

The efficiency of an air-lift pump is generally measured by the ratio of the water horse-power in useful work done to the air horse-power supplied at the foot-piece. The over-all efficiency of the plant, however, is the ratio of the water horse-power raised to the I.H.P. at the compressor or to the

* *Proc. Inst. Mech. Engineers*, Nov., 1917, p. 613.
† *Bulletin*, University of Wisconsin, U.S.A., No. 667, 1911.

work put into the compressor. In general practice it appears that efficiencies of $\frac{\text{water horse-power}}{\text{air horse-power}}$ average about 40 per cent, although the experiments of both Dr. Owens and Mr. Purchas show quite conclusively that 60 per cent may be achieved by careful study of the best proportions for each set of conditions. The over-all plant efficiency depends considerably upon the compressor, and may reach 30 to 35 per cent.

In this country many important air-lift pumping schemes have been installed by Messrs. C. Isler & Co., Messrs. Le Grand & Sutcliffe, and by the Worthington Pump Company, all of London.

FANS AND AIR COMPRESSORS

BY

J. E. ELLOR

Fans and Air Compressors

The several types of fan and air compressor, commonly used in practice, work on one or the other of two fundamental principles, namely, *direct compression* of the air in a cylinder, or by imparting *momentum or kinetic energy* to the air by means of a rotating impeller, and afterwards converting the velocity head back to pressure head before final discharge.

Direct Compression.—Air being an elastic fluid, the work expended on it by decreasing its volume creates increases in pressure, temperature, and density which follow definite laws. By this principle of compression high pressures of delivery may be obtained, but the flow of air is intermittent on account of the non-delivery period of charging and compressing.

Compression on the Momentum Principle.—If work is done on a column of moving air free to expand (i.e. an open discharge), then momentum or kinetic energy is imparted to that column, thus increasing its velocity. To convert the velocity into pressure at the final discharge, the stream or column of air must be allowed to expand in sectional area along the line of flow. This expansion has the effect of increasing the pressure at the expense of velocity. With this principle only relatively small pressures can be obtained over any one stage, but large volumes of air are dealt with, and the method has the advantage of producing continuous flow.

Air Data.—

1. The specific volume of air before entering the machine is a function of its temperature, barometric pressure, and humidity.

2. *Dry Air* is defined as air containing no moisture.

3. *Saturated Air* is defined as air having in suspension the maximum amount of moisture.

4. Temperature and barometric pressure change with weather conditions and altitude.

Values of atmospheric pressure and temperature at various altitudes are given in Table I.

5. *Free Air.*—For air at normal sea-level conditions, referred to as *free air*, we have the following constants and equivalents.

(*a*) Normal atmospheric pressure = 14·7 lb. per square inch ≡ 29·9 in. of mercury ≡ 33·94 ft. of water.

(*b*) Density or weight per unit volume is 0·076 lb. per cubic foot.

(*c*) Specific volume, i.e. $\frac{1}{\text{density}}$, is 13·15 c. ft. per pound.

(*d*) Temperature is 62° F.

These conditions are often abbreviated by referring to air at N.T.P. (normal temperature and pressure, i.e. 62° F. and 14·7 lb. per square inch).

Pressure Equivalents:

1 lb. per square inch ≡ 2·04 in. of mercury ≡ 2·309 feet of water.
1 in. of mercury ≡ 0·49 lb. per square inch ≡ 1·132 feet of water.
1 ft. of water ≡ 0·433 lb. per square inch ≡ 0·883 in. of mercury.

TABLE I

Altitude, Feet.	Pressure, Pounds per Square Inch Absolute.	Temperature, Degrees F.	Density, Pounds per Cubic Foot.
0	14·7	62	0·076
2,000	14·17	52	0·0717
4,000	13·17	44·6	0·0675
6,000	11·79	37·4	0·0635
8,000	10·92	30·0	0·0597
10,000	10·1	23·0	0·0561
15,000	8·3	16·0	0·0479
20,000	6·76	12·0	0·0406

Formulæ applied to a Moving Column of Air.—

Notation:

H is the head of air in feet;
h, the velocity head in inches of water;
h_i, impact head in inches of water, i.e. the pressure head + frictional resistance + velocity head;
v, velocity in feet per second;
V, velocity in feet per minute;
W, weight of air in pounds per cubic foot;
P, pressure in pounds per square foot;
Q, delivery in cubic feet per minute;
B, barometric pressure in inches of mercury;
t, temperature in degrees F.

If H is the equivalent height in feet of a column of air of the same density necessary to produce a pressure at its base equal to h inches of water,

$$\mathrm{WH} = 5{\cdot}2h.$$

$$\text{But } \mathrm{W} = \frac{1{\cdot}328\mathrm{B}}{(460 + t)}.$$

$$\therefore \mathrm{H} = 3{\cdot}91(460 + t)\frac{h}{\mathrm{B}}.$$

Velocity due to a Given Head.—The velocity acquired by air in moving under a given pressure or head is exactly the same as that acquired by a stone

in falling freely from a given height; i.e. if v is the velocity in feet per second and g the acceleration due to gravity in feet per second per second,

$$\frac{v^2}{2g} = \mathrm{H},$$

where H is the head of air in feet.

$$\therefore v = \sqrt{2g\mathrm{H}},$$

$$\text{and } \mathrm{WH} = 5{\cdot}2h.$$

$$\therefore v = \sqrt{\frac{2g \times 5{\cdot}2h}{\mathrm{W}}},$$

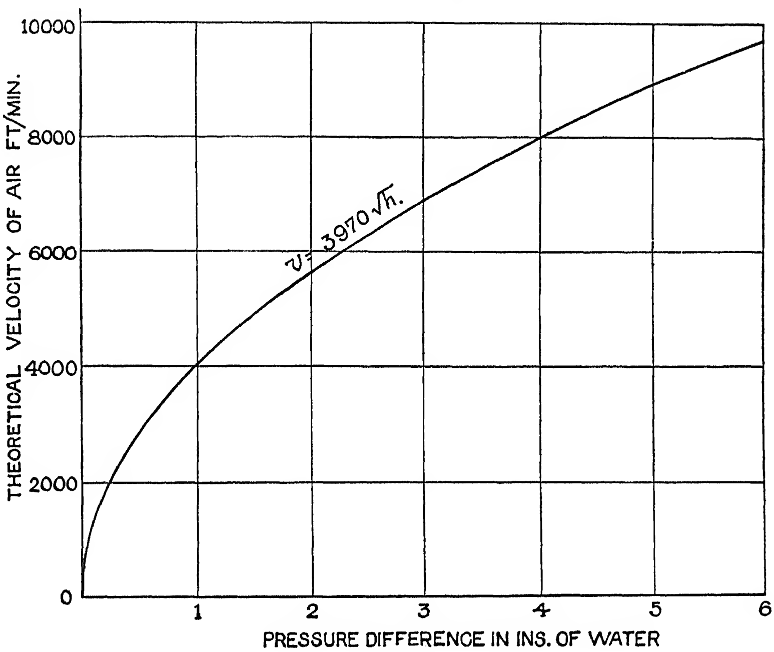


Fig. 1

i.e. $\mathrm{V} = 3970\sqrt{h}$ on reducing the velocity to (ft./min.) units. h is the *velocity head* in inches of water. Values of V corresponding to values of h up to 6 in. are plotted in fig. 1.

For other conditions of temperature and barometric pressure

$$v = 15{\cdot}9\sqrt{(460 + t)\frac{h}{\mathrm{B}}} \text{ (feet per second).}$$

$$\text{Pressure P} = 5{\cdot}2h_i = \mathrm{HW}.$$

$$1 \text{ oz. per square inch of pressure} \equiv 1{\cdot}732 \text{ in. of water.}$$

$$\text{Air horse-power} = \frac{\mathrm{VAP}}{33000} = \frac{\mathrm{QP}}{33000} = \frac{5{\cdot}2\mathrm{Q}h_i}{33000}.$$

(A = area of outlet in square feet.)

Fundamental Laws governing Air Compression.—When air is compressed on either of the two principles, the work done on it during compression is absorbed partly in increasing the pressure energy and partly in increasing the heat energy. Compression of air always develops heat on account of the molecular activity of the particles, which shows itself in rise in temperature.

With rise in temperature the air expands in volume, and, conversely, the lower the temperature the smaller will be the volume. The weight per unit of space occupied is lower at high temperatures than at low for the same pressure; i.e. by reducing temperature the density increases.

To obtain the greatest volume of air compressed to a given pressure, it is first necessary to reduce the intake temperature as low as possible, and to maintain a minimum temperature during compression in order that its final density may be high.

This explains the desirability of cooling, by water jacketing, the compressing chambers, and employing intercoolers between the stages when high pressures are required.

A certain amount of moisture is always present in atmospheric air, and any increase in temperature raises its moisture-absorbing capacity. From this it follows that the cooling of air, whilst reducing volume, also causes the compressed air to give up a portion of the moisture contained. Hence it is necessary to provide a moisture trap, either in the intercooler or at the receiver at the final discharge.

There are two ways in which air may be compressed or expanded:

An *adiabatic expansion* (*or compression*), in which heat is not allowed to enter or to leave the air.

An *isothermal expansion* (*or compression*), in which the *temperature* of the air does not change during the change of state.

Definite relations connect the pressure, volume, and temperature of dry air.

Representing the respective conditions by p, V, and T,

where p is in pounds per square inch absolute,
V is the volume in cubic feet,
T is the absolute temperature in degrees F., i.e. $(460 + t)$,

also letting suffix o refer to the initial conditions, and letting suffix 1 refer to the final conditions or intermediate points, we have for *adiabatic changes*:

$$p_o V_o^{1\cdot41} = p_1 V_1^{1\cdot41} \text{ and } \frac{p_o V_o}{T_o} = \frac{p_1 V_1}{T_1} \text{ (the gas equation);}$$

or, expressed as ratios,

$$\frac{p_1}{p_o} = \left(\frac{V_o}{V_1}\right)^{1\cdot41} = \left(\frac{T_1}{T_o}\right)^{3\cdot46}; \quad \frac{V_o}{V_1} = \left(\frac{p_1}{p_o}\right)^{\cdot71} = \left(\frac{T_1}{T_o}\right)^{2\cdot6};$$

$$\frac{T_1}{T_o} = \left(\frac{V_o}{V_1}\right)^{\cdot41} = \left(\frac{p_1}{p_o}\right)^{\cdot29}$$

The theoretical horse-power required to adiabatically compress and deliver a volume of air against a maintained pressure is:

$$\text{Air horse-power} = 0{\cdot}015 p_o V_o \left[\left(\frac{p_1}{p_o} \right)^{\cdot 29} - 1 \right],$$

where V_o is the delivery of air at inlet in cubic feet per minute.

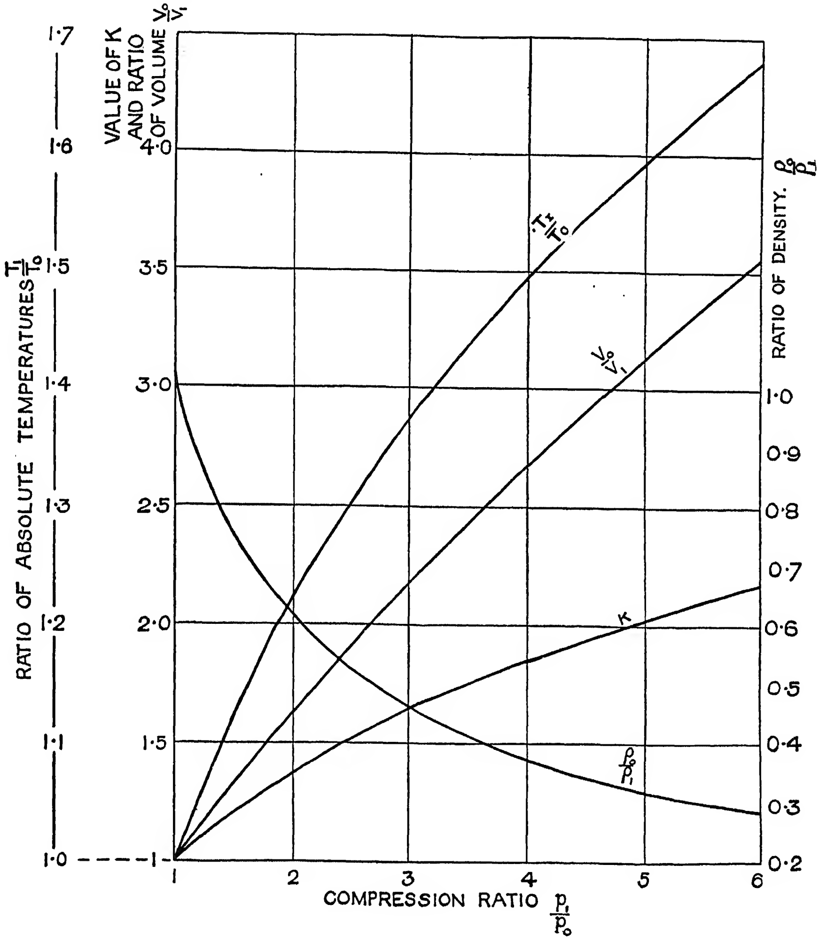


Fig. 2

Curves shown in fig. 2 give ratios of volumes, absolute temperature, and density for compression ratios up to $\frac{p_1}{p_o} = 6$. They may be applied to any condition of the air surrounding the inlet, whether on the ground level or at an altitude.

The "K" curve gives the adiabatic air horse-power by taking:

$$\text{Air H.P.} = K \times \frac{p_o V_o}{230}.$$

For *isothermal changes* we have:

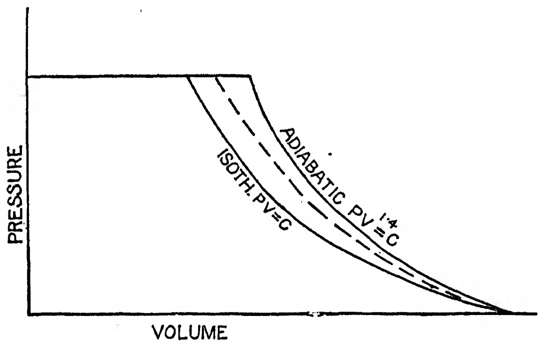
$$p_o V_o = p_1 V_1,$$

$$\text{i.e. } p_1 = p_o \frac{V_o}{V_1},$$

$$\text{and air horse-power} = 0{\cdot}00436 p_o V_o \left(\log_e \frac{p_1}{p_o}\right),$$

$$\text{where } \log_e \frac{p_1}{p_o} = 2{\cdot}3026 \times \log_{10} \frac{p_1}{p_o}.$$

A graphical representation of the two laws of compression is shown in fig. 3, where pressures are plotted against volumes. The areas of each diagram are proportional to the air horse-power.



To compress equal volumes of air to a given pressure requires less work *isothermally* than *adiabatically*.

It is, however, impossible to compress along the isothermal line in practice, on account of the slow rate at which the heat may be dissipated during any one act or stage of compression. With the water-jacket cooling systems the compression curve usually lies between the adiabatic and the isothermal line, shown dotted.

By cooling the compressed air between the stages of compression in an intercooler, a description of which is given later, a greater amount of heat is dissipated over the whole or total range of compression, hence the isothermal is more closely approached.

A pressure-volume diagram for two-stage compression is shown in fig. 4. There is a slight pressure drop between the exit from one cylinder and the inlet to the other owing to the frictional resistance of valves and cooler passages.

Centrifugal Fans.—Definite rules or formulæ for the determination of the sizes of fans cannot be stated, as there are so many variable factors in the design of each type, such as the shape of wheel, blades, volute casing, &c.

Guided by fundamental formulæ and analysis of tests on known types, manufacturers produce new fans for any particular duty.

Usually the head against which a fan operates is that due to the frictional resistance in the ducts or pipes. Where small pressures are required at the discharge, the head, or *maintained resistance*, is the sum of frictional resistance and static pressure.

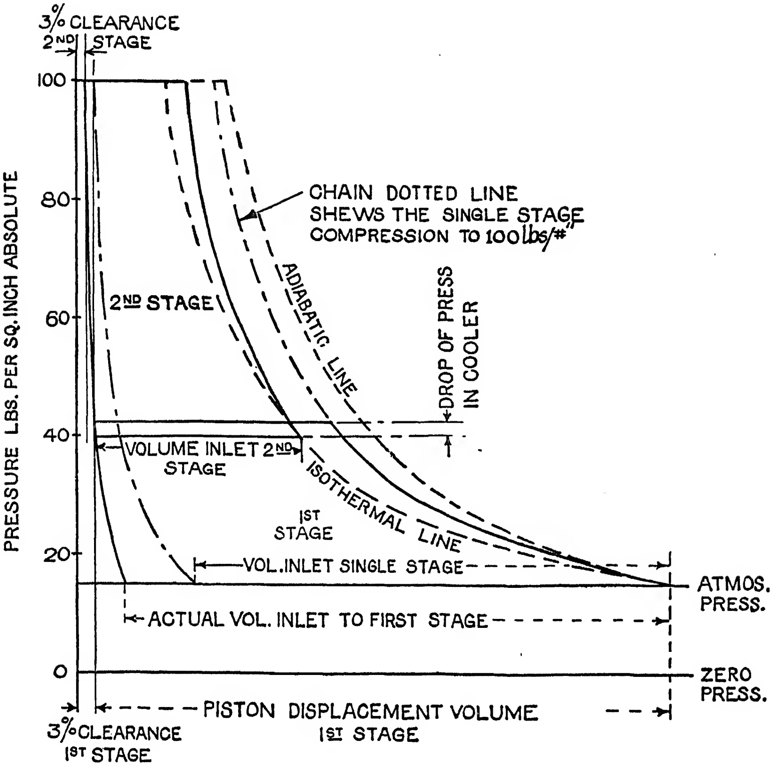


Fig. 4

The *impact pressure* is the total equivalent head that the fan gives to the st, and equals the sum of the maintained resistance and head due to the velocity at discharge. ɔm the impact pressure : *air horse-power* is com- :ed.

Mechanical efficiency is the io of air horse-power to wer required to drive the ., which includes mechan- l friction. Efficiencies vary ;htly with the type of fan; ɔommon figure attained is the order of 50 per cent. ;. 5 shows the character- c curves of a multi-blade . having sixty blades of all radial depth, forward :ved angle, and plane sur- e, having a tip speed of

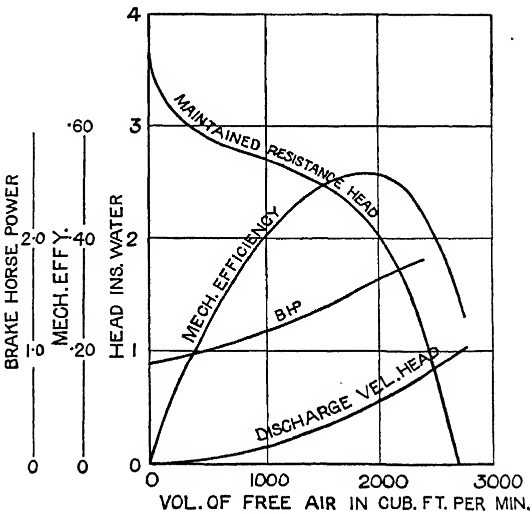


Fig. 5

5000 ft. per minute and discharging into a volute casing. Results of tests for each type of fan plotted in this manner will furnish their own particular characteristics.

From such curves the performance at any other peripheral speed may be determined for the same type of fan by applying the following laws:

The volume dealt with varies directly as the peripheral speed; the head, or maintained resistance, as the square of the speed; power as the cube of the speed.

The efficiency will remain constant for any particular load-point on the curve.

For a given load, at a given speed, the capacity will vary as the square of the diameter of the fan wheel.

Formulating the above laws for any given fan whose wheel diameter is D ft.,

$$\text{The peripheral speed} = 3{\cdot}14DN \text{ ft. per minute.}$$

$$V \propto N; \; h \propto N^2; \; \text{B.H.P.} \propto N^3.$$

Change of initial density or air pressure will effect the head and horse-power in direct proportion.

From fig. 5 it will be seen that the fan works at its highest efficiency over a small range of capacity, and suffers in performance when working either below or beyond its capacity or full blast. When working below its capacity, air is slipping back between the blades at some points and delivering at others.

Fans having radial corrugations in the blades maintain a high efficiency over a greater range of capacity.

The chief features affecting the performance are:

1. Size and shape of air entrance.
2. Relative size, length, depth, curvature and angles, and number of blades.
3. The casing in which the wheel is placed.

The diameter and width of blades depend largely upon the quantity passing. The inlet diameter is usually a function of the wheel diameter, and the radial velocity at the blade entrance ought to be approximately that of the flow through the fan inlet.

The radial area at blade entrance ought then to be equal to the smallest area of the fan inlet.

The head against which the fan operates is a function of the peripheral speed, i.e. wheel diameter and rotational speed. Theoretically, when any centrifugal impeller is delivering its full blast, the pressure head when leaving the tips is equal to the velocity head, or, in other words, the total head at the wheel tip exists as half pressure and half velocity energy.

Fan Wheels.—Losses common to all fans occur when imparting momentum to the air during its passage through the wheel, due to surface friction and sudden changes of velocity and direction. To minimize these

losses, the entrance and exit angle of the blades must be chosen to allow the air to enter and leave without shock.

Referring to figs. 6 to 8, representing the diagram of velocities where suffix 1 and 2 are the entrance and exit conditions to the wheel,

> α is the angle the blade makes with the tangent;
> u, peripheral velocity;
> w, the actual velocity through the blade channels;
> v, the actual velocity relative to a stationary point;
> v_r, the radial component of velocity.

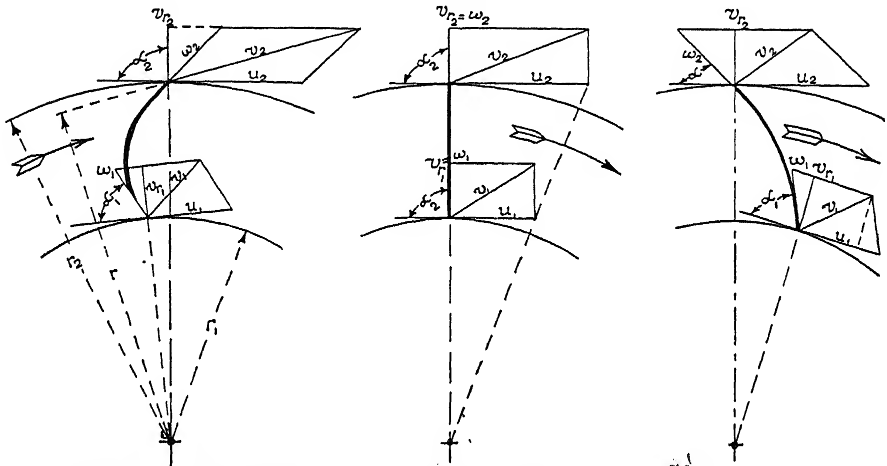


Fig. 6 Fig. 7 Fig. 8

The velocity diagrams are for fans having the air inlet at the centre and axial flow at right angles to the wheel. v_r then is usually equal to the velocity of flow at the eye, and the width is chosen to pass the necessary quantity.

Loss of head at blade entrance is $(u_1 - v_{r_1}) \dfrac{\cot\alpha_1}{2g}$.

A minimum loss of head occurs when $\dfrac{u_1}{v_{r_1}} = \cot\alpha_1$.

The loss of head at exit, if discharged into the atmosphere, is $\dfrac{v_2^2}{2g}$; if into a diffuser or volute casing, part of the velocity v_2 is converted into pressure.

The three diagrams show blades having their generatrices parallel with the axis, curved forward, radial, and backwards respectively. Comparing the magnitudes of the exit velocity v_2, the forward-curved blades will produce the greatest head for equal tip speeds. Where forward-curved blades are used, the wheel should discharge into a volute or spiral casing. Where discharging direct into the atmosphere or into an open diffuser, backward-curved blades are best.

In any fan wheel there is a tendency toward uneven flow at the periphery

due to change of axial flow at the inlet to radial flow at the discharge. The air tends to creep along the blade axially and bank up at the side remote from the inlet. To counteract this, either the inlet extremities of the blades are arranged to lag behind the delivery extremities in the direction of rotation, or the diameter at the blades nearest the inlet side should be slightly greater and gradually decrease down to the diameter of the wheel-disc side.

A further point is the taper of the inlet edge of blades parallel with the axis; this adds to the efficiency of air entrance.

For particular duties, other types of wheels are made having the shape of blade at the inlet in the form of a screw propeller, finishing at the

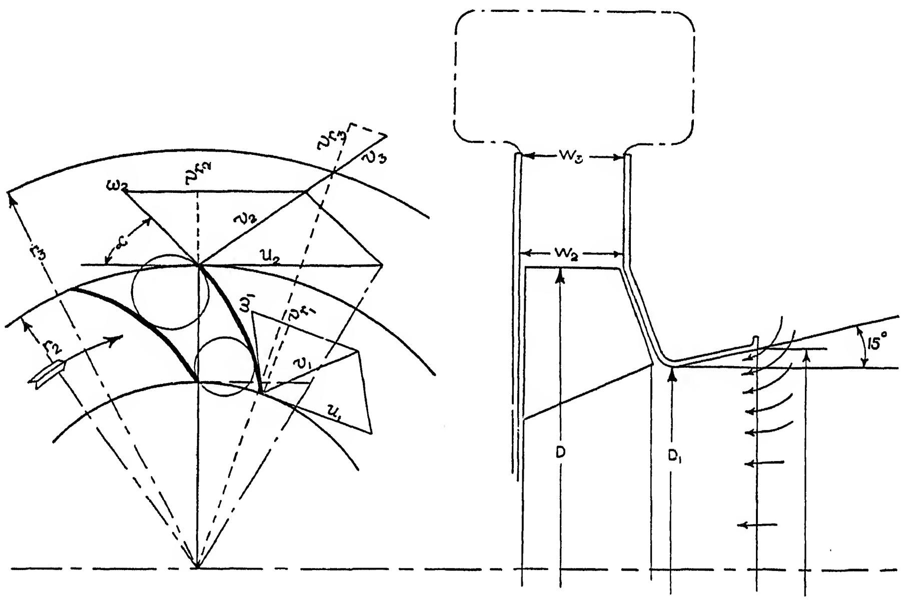


Fig. 9

periphery as a plain blade; also the screw-propeller fan is commonly used when large volumes of air are to be displaced against no pressure. In this particular design the air flows axially, varying in velocity radially along the blade. Its action is similar to that of the air screw for an aeroplane.

Fan Casings.—It is usual to express the principal dimensions in terms of the wheel diameter.

Let D be the diameter of the wheel and D_1 the smallest diameter of the fan inlet. The capacity is influenced both by blade area or surface and the restriction to flow at the inlet. A good compromise is to make D_1 not less than 0·625D. Also, it is better to attach a converging conical entrance piece, as shown in fig. 9.

Two inlets are sometimes employed where large volumes are dealt with, in which case the impeller has blades attached to either side of the wheel disc.

The *wheel discharge chamber* is often referred to as the *diffuser* or *volute casing*.

When a fan is fully loaded, i.e. delivering a full blast, air is given off at a uniform rate per unit length of periphery of wheel. There must therefore be a uniform increase in area outside the wheel for the passage of air.

Two types of housings are commonly employed.

Arrangement I.—An annular space, having a free outlet all round the periphery, surrounds the wheel. In this type of casing, known as a *diffuser casing*, the peripheral area increases with the radius. This has the effect of reducing the air velocity and converting it into pressure. Also, the surrounding pressure acts radially, and in order that the direction of flow, when leaving the blades, shall be as near radial as possible, backward curvature should be employed.

The diameter of the diffuser chamber is found from the velocity diagram (fig. 9), if a final discharge velocity is assumed (usually about 500 to 1000 ft. per minute).

The volume passing through is known, and the products of the radial velocity v_{r_3} and peripheral areas must be equal at any part.

We have then the following equations:

$$v_{r_2} \times A_2 = v_{r_3} \times A_3,$$

$$\frac{A_2}{A_3} = \frac{W_2 r_2}{W_3 r_3},$$

where W_2 is the width of diffuser chamber at entrance, and W_3 is the width of diffuser chamber at exit.

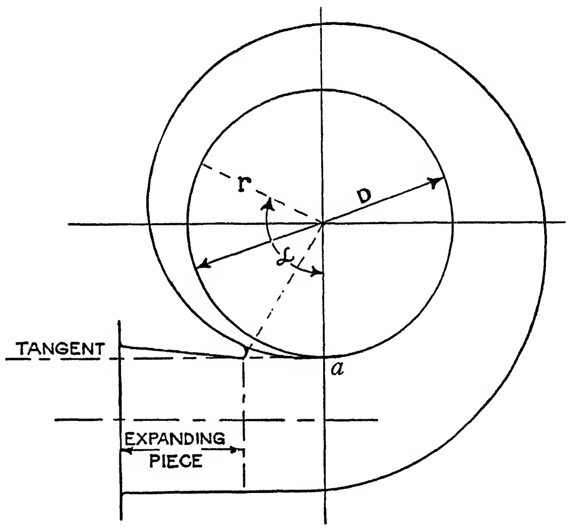


Fig. 10

The width at entrance is usually that of the wheel, and the sides may be radial or slightly divergent or convergent. Change in width will have an effect upon the radius r_3 for a given discharge velocity. In the case of a divergent casing, the angle must not exceed 7° taper.

Arrangement II, Spiral or Volute Casing.—With this type of housing the air is collected from the periphery of the wheel before discharge. Therefore the airway must be of increasing area. The change in area is usually affected by change of curvature of the outer wall by making it of spiral form.

Fig. 10 shows the general shape of a volute or spiral casing.

The point *a* is known as the *cut-off point*, where the spiral discontinues its approach to the wheel circumference. On reaching this point, after one revolution, the wheel has delivered the whole of its contents.

Commencing the spiral from the point of intersection with the wheel circumference at the tangent drawn parallel to the axis of discharge, its

equation will be $R = r + k\alpha$, where R is the distance from wheel centre to a point on curve corresponding to angle α measured from point of intersection. For the common form of multi-blade fan, $R = r + 0{\cdot}2\alpha$, where α is in radians.

In practice the *cut-off* point is moved back about 30° to allow a little clearance and to permit the nose to be rounded off. To do this adds sensibly to the prevention of noise.

This leaves a small portion of the wheel periphery exposed or uncovered by the spiral, and on this section there will be a tendency for slip or air leak, back through the fan, due to the pressure at the tip being slightly less than at the point of cut-off. The practical advantages, however, by retarding the *cut-off* point warrant the small amount of slippage.

In the case of a fan having two discharge branches the same spiral may be used, stopping off at 180°, the second being started from this point. The linear dimensions and areas of volute will be approximately halved.

Fans having high tip speeds with forward-curved blades sometimes have a diffuser casing and volute when high efficiency is required; also a more complete conversion of velocity into pressure is effected with the help of a diverging pipe from the point of cut-off at the volute discharge.

RECIPROCATING COMPRESSORS

In the piston type of compressor the work of compression is done intermittently, one complete cycle being performed per revolution when single acting and two when double acting. A volume of air is drawn through automatic or mechanically operated inlet valves into the cylinder as the piston moves outwards; compression and delivery through other similar valves takes place on the return stroke.

Piston displacement is the volume swept by the piston on the suction stroke. Although not a true measure of the capacity, it forms the base from which the capacities of compressors are compared.

Delivered capacity is the actual volume of free air (at atmospheric pressure) delivered.

True volumetric efficiency is the ratio of the volume of free air drawn in to the piston-displacement volume. This ratio depends largely upon the clearance space and efficiency of valves.

The *clearance space* is the total space existing between the piston crown, when at top dead centre, and the cylinder head, including valve ports. Its effect on volumetric efficiency is very important, owing to the fact that a maximum pressure remains in the enclosed space after valves close. As the piston recedes the volume of compressed air expands approximately isothermally until the pressure falls below that necessary to induce the flow of a fresh charge. Referring to fig. 4, it will be seen that a considerable portion of the piston displacement is thus robbed of its capacity to charge. A common figure for clearance volume is about 3 per cent of the displacement for small machines, and about 1 per cent for large machines.

The *apparent volumetric efficiency* is represented by the volume up to the point where the expansion line meets the suction-pressure line, divided by the piston-displacement volume. It differs from the true efficiency in that the average condition of temperature and pressure in the cylinder at the end of the suction stroke have changed relatively to those of the outside air.

The *mean effective pressure* is the average pressure attained throughout the two strokes or per cycle, usually expressed in pounds per square inch units.

For compression in one cylinder $p_m = \frac{n}{n-1}p_o\left[\left(\frac{p_1}{p_o}\right)^{\frac{n-1}{n}} - 1\right];$

for multi-stage compression $p_m = \frac{sn}{n-1}p_o\left[\left(\frac{p_1}{p_o}\right)^{\frac{n-1}{ns}} - 1\right];$

where s is the number of stages or cylinders, n the index of compression which varies from 1·3 to 1·35, depending upon cylinder design and means of cooling, p_1 is the final pressure, and p_o the initial pressure in *each* cylinder in pounds per square inch.

Single-stage compressors are built for maximum pressures up to 100 lb. per square inch; two-stage compressors up to 500 lb. per square inch; three-stage compressors up to 1000 lb. per square inch.

Between each stage it is necessary to pass the air through an intercooler in order to extract the heat of compression generated in the preceding cylinder.

With perfect intercooling, and neglecting the clearance volume, the theoretical horse-powers required to compress air are as follows:

Single stage:

$$\text{H.P.} = \frac{144}{33000}\,\frac{n}{n-1}p_o V_o\left[\left(\frac{p_1}{p_o}\right)^{\frac{n-1}{n}} - 1\right];$$

Two stage:

$$\text{H.P.} = 2 \times \frac{144}{33000} \times \frac{n}{n-1}p_o V_o\left[\left(\frac{p_1}{p_o}\right)^{\frac{n-1}{2n}} - 1\right];$$

Three stage:

$$\text{H.P.} = 3 \times \frac{144}{33000} \times \frac{n}{n-1}p_o V_o\left[\left(\frac{p_1}{p_o}\right)^{\frac{n-1}{3n}} - 1\right];$$

where p_o, V_o are initial pressure and volume, and p_1 is final pressure, the units being cubic feet for volume and pounds per square inch for pressure.

The intercooler pressure of the two-stage compressor .. $= \sqrt{p_o p_1}$,

The intercooler pressure of the three-stage compressor .. $= \sqrt[3]{p_o^2 p_1}$ for the first,

and $= \sqrt[3]{p_o p_1^2}$ for the second.

With perfect intercooling, then, the size or volume of each cylinder should be in inverse proportion to pressure at admission.

In actual practice the temperature, after passing an intercooler, is slightly higher than the initial temperature of the free air, whilst the exit pressure has fallen below that of the inlet. The volume of air delivered by a given compressor at altitude a, compared to that delivered at sea-level, is given by the equation:

$$\frac{V_a}{V_o} = \frac{1 + \frac{P}{p_o}}{1 + \frac{P}{p_a}},$$

where P is the gauge pressure in pounds per square inch, and p the barometric pressure in pounds per square inch.

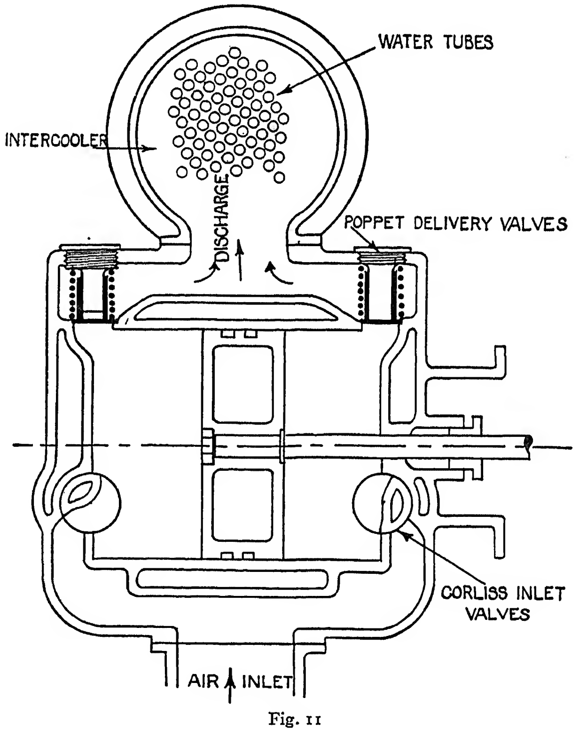


Fig. 11

Although the volume delivered decreases with altitude, the power does not decrease in the same proportion. The theoretical power may be computed from the horse-power formulæ by substituting $p_a V_a$ for $p_o V_o$, the ratio of compression remaining the same for a given machine.

Cooling.—Injecting water in the form of a spray has been tried as a means of cooling, and compression curves of $pV^{1 \cdot 2}$ have been obtained. The chief objection to this so-called " wet compression " is the reduced capacity consequent upon the moisture affecting the initial density, and the mechanical auxiliaries. Further, because of the time element required to vaporize and

mix thoroughly with the incoming air, such machines will not operate efficiently at high speeds.

External cooling is almost exclusively adopted. For the smaller sizes of cylinders the outside walls are ribbed, and radiate the heat to the surrounding atmosphere. The larger sizes are water-jacketed, giving an average compression $PV^{1 \cdot 35}$. The quantity of water necessary is a function of the inlet temperature, a common figure being *twice the inlet temperature in degrees F. equals the number of gallons per hour per* 100 *cubic feet per minute.*

Intercoolers consist of long cylindrical chambers fitted with longitudinal tubes through which water is circulated, the compressed air to be cooled passing around and along the circumference. The flow of water and air should be in opposite directions, in order that the coolest air may meet the coolest water. A section of a cooler is shown mounted on a compressor in fig. 11.

The number of cubic feet of free air per minute per square foot of cooling surface is given as $0{\cdot}4\ (t_a - t_w)$, where t_a is the temperature of air leaving, t_w that of the water entering, both in degrees F.

Valves.—For high volumetric efficiency, the inlet valves should have a large area of passage with rapid opening and closing to prevent wire-drawing of the air. It is also very important that the delivery valves should close quickly, as the leakage back into the cylinder reduces both capacity and efficiency.

Combination of Valves.—The most desirable combination of inlet and delivery valves would be to have the inlet valve a mechanically operated one, such as a Corliss, and for the delivery valve to open automatically and close mechanically.

In many machines a number of small *poppet valves* working automatically are employed both for inlet and delivery. These are fairly efficient and reliable, but are somewhat noisy in operation. Another good type commonly used is the *plate valve*, made of thin tempered steel or bronze. These valves are very light, and consequently the inertia forces are small, thus facilitating rapid opening and closing.

The area through the inlet valves should not be less than 10 per cent of the piston area, and for the delivery 12 per cent. These percentages apply to normal piston speeds of 400 ft. per minute. Higher piston speeds of 800 ft. per minute are adopted, in which case the valve areas should be in the order of 13 per cent and 15 per cent.

Figures of Efficiencies.—The true volumetric efficiency varies from 80 to 96 per cent.

The *cylinder efficiency* is the work required to compress isothermally a volume of air equal to the piston displacement, disregarding clearance, divided by the actual work done as measured from an indicator diagram. It varies from 80 to 78 per cent.

The *efficiency of compression* is the volumetric efficiency multiplied by the cylinder efficiency, so that the range will be from 60 to 82 per cent.

The *over-all efficiency* is the efficiency of compression multiplied by the

mechanical efficiency, the latter varying between 80 and 95 per cent, making an over-all efficiency of 48 to 78 per cent.

Turbo-blowers and Compressors.—The construction of turbo-blowers and compressors is practically identical, the term *blower* being applied to machines for pressures up to 15 lb. per square inch gauge, and *compressor* to those for pressures above 15 and up to 100 lb. per square inch. The final delivery pressure is a function of the number of wheels and of the peripheral velocity. Where the air to be compressed is at a low initial density, the relative number of wheels must be increased. When dealing with large volumes of air the pressure per wheel should not exceed 5 lb. per square inch, and the diameters should be chosen as large as the centrifugal stresses will permit, in order to keep the speed of rotation relatively low. In this type of compressor, the velocities being high over a large passage surface, the work lost in air friction is fairly great, and varies as the square of the velocity, hence it is advantageous to keep down the peripheral speeds.

Where lightness of unit is of more importance than efficiency, small diameter single-wheel compressors are employed, rotating at speeds up to 30,000 r.p.m. Where many wheels are used, special means of cooling the air during its passage from one wheel to another are sometimes provided by arranging water jackets around the casings. For pressures up to 100 lb. per square inch two sets of wheels are placed on the same driving shaft, and a middle bearing placed between the two casings. The air passes through an intercooler on its way from the last wheel of the first set to the first wheel of the second set.

Turbo-compressors occupy comparatively little space, the construction is simple, and, with the exception of the bearings, there are no metal-to-metal rubbing surfaces. Also, when carefully balanced, vibrations do not exist. A constant delivery pressure can be obtained over quite a large range of volume, and, where constant volume discharge is necessary, a governor, operated by the air velocity at discharge, restricts the outlet. These types of machines are useful in connection with exhausters, blast furnaces, and oil burning.

Principle of Working.—Very much the same action takes place as occurs in the centrifugal fan. The characteristic effects of shape of blades are the same, but the wheel tip speeds are higher, and special attention is paid to the means of converting the velocity head of the air into pressure. Fixed vanes are placed in the *diffuser space* around the wheel periphery at an angle corresponding to that of exit velocity V_2, and the areas of both the *impeller blade channels* and *diffuser channels* must allow for change of density with increase of pressure and temperature.

Fig. 12 shows a sectional arrangement of a three-stage machine compressing up to 10 lb. per square inch gauge, above which is a curve of pressures and temperatures corresponding to the inlet and exit of each wheel. Air enters the eye of No. 1 impeller at A, and is delivered at the periphery B. The fixed diffuser vanes are placed between B and C, which collect the air

at high velocity. The air attains a maximum pressure at D in a well-designed machine. *Guide blades* are inserted in the passage from D to E, to steady the stream of air and direct it into wheel No. 2, where the operation is repeated.

It will be noted that, as the number of stages increases, the densities of the compressed air at the inlet and tip of each impeller increases with the

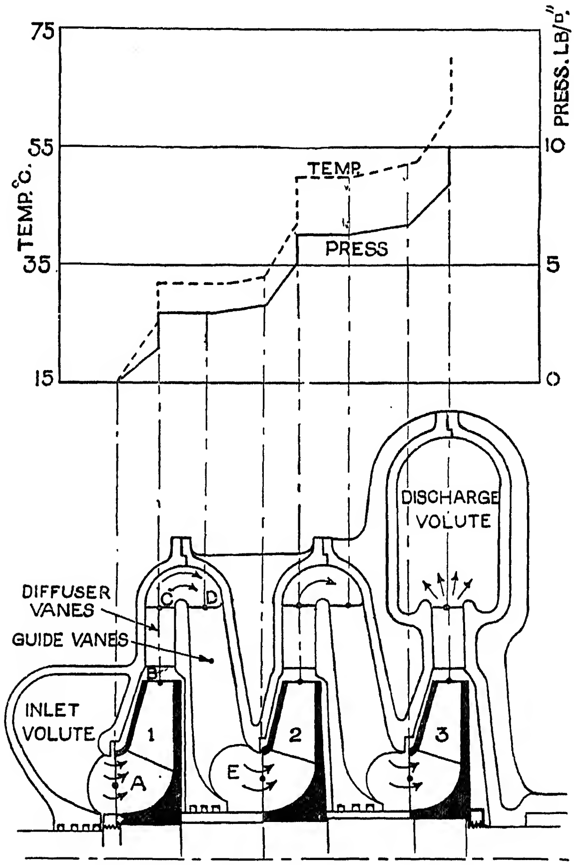


Fig. 12

pressure; hence, for the same diameter of wheels, the axial width of blade will decrease in proportion to the densities.

Constant speed characteristic curves are shown in fig. 13. The performance at any other speed may be determined by proportioning the points on a known curve at a speed N_1 in the following manner:

$$Q_2 = Q_1 \times \frac{N_2}{N_1},\ p_2 = p_1 \left(\frac{N_2}{N_1}\right)^2,$$

where Q is the quantity of free air in cubic feet per minute and p is the pressure in pounds per square inch.

The efficiency at the relative points on each curve will be approximately the same, so that $(B.H.P.)_2 = (B.H.P.)_1 \times \left(\frac{N_2}{N_1}\right)^3$.

As the quantity passing on the constant-speed curve is reduced, the pressure rises up to a maximum, and, if further reduced, violent oscillations

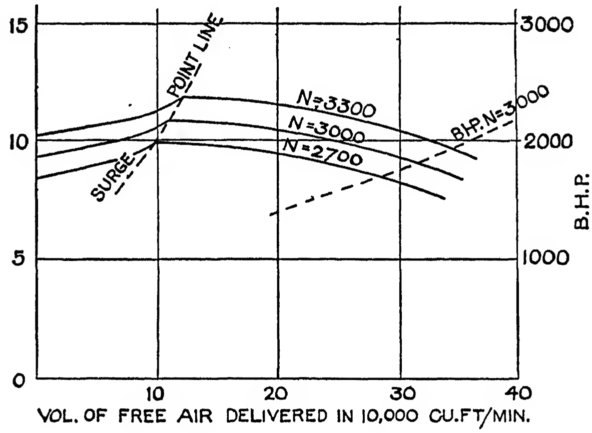


Fig. 13

are set up, resulting in unsteady conditions. Where these conditions occur is known as the *surging point*.

Impellers.—Fig. 14 shows the blades of an impeller, radial at the exit, and inclined to receive the air at the inlet for minimum loss of head. The

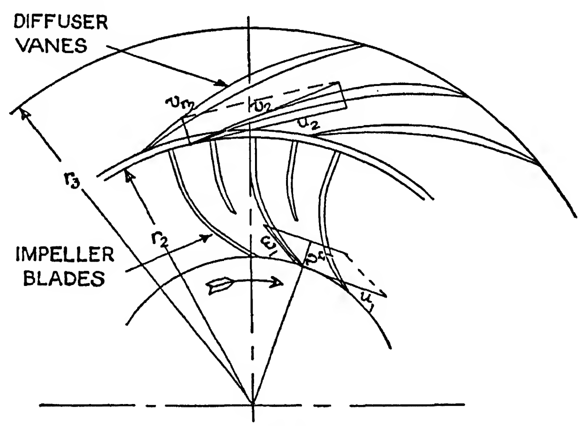


Fig. 14

velocity diagrams referred to in the case of fan design apply also to this type of compressor, and the effect of backward and forward inclination of blades has an important bearing upon the pressure rise for equal peripheral speeds. The channels formed by the blades should not be allowed to expand too rapidly in area, and it is found necessary to insert half blades in

many cases. In order to keep down the frictional losses the number of blades employed varies from sixteen to twenty, and, when choosing the areas of channels, the change in density of the air whilst passing from inlet to exit should be taken into account.

The total rise of pressure per stage is given by the following formula:

$$p_1 - p_o = \frac{72\rho_m u_2^2}{g},$$

where ρ_m is the mean density in pounds per cubic foot;
u_2, peripheral velocity of wheel;
g, 32·2 ft. per second per second;
p, the absolute pressure in pounds per square inch.

$$\text{Compression ratio, } r \text{, is } \left\{\frac{72\rho_m u_2^2}{gp_o} + 1\right\}.$$

Total compression ratio in multi-stage compressor, R, equals r^n, where n is the number of wheels.

Diffuser Vanes.—It is usual to have a smaller number of diffuser vanes than impeller vanes, and the inclination must coincide with the wheel exit velocity v_2, from which, together with the density, the entrance areas are chosen. The rate of expansion or taper of channels must not exceed 7°.

Multi-stage compressors must have some means provided for balancing the thrust caused by the pressure difference between the inlet and back side of each wheel. This is accomplished either by balance pistons on the end of the shaft, or by opposing the impellers in two groups.

Efficiencies are referred to in two ways.

$$\textit{Relative to adiabatic compression} = \frac{\text{adiabatic air temperature rise}}{\text{actual air temperature rise}}.$$

$$\textit{Over-all efficiency} = \frac{\text{air horse-power}}{\text{power required to drive}}.$$

The former is usually about 75 per cent, and the latter varies between 60 and 70 per cent.

Jet Blowers and Compressors.—This type of compressor also works on *the momentum principle*, but, instead of a centrifugal fan supplying the energy, a jet of steam or water issuing from a nozzle at a high velocity imparts a portion of its momentum to the air in contact with the stream. Fig. 15 shows diagrammatically the construction of such an apparatus. Under the influence of pressure the medium leaves the small nozzle A, and the ejector action induces air or gas to flow into the converging nozzles C. The air velocities are a maximum in any one nozzle at the exit. At the exit of the final converging nozzle D, a tail pipe or diverging cone is attached, in order to convert the velocity at the inlet into pressure at the outlet.

The number of nozzles employed and their relative areas are largely a

matter of experiment, depending upon the capacities and degree of pressure required. For simplicity and reliability they are far away the best, but the efficiency of working is exceedingly low. They are particularly useful in producing a vacuum, and are often employed as exhausters as well as in

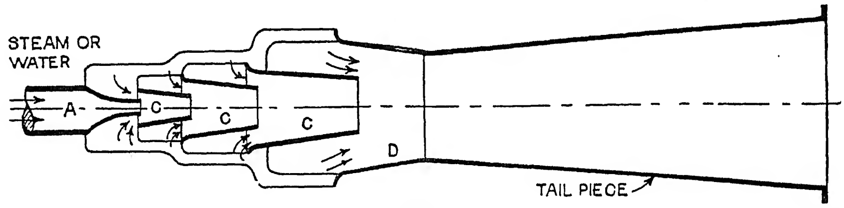


Fig. 15

connection with condensing plants for ridding the system of air. In the latter case they must be capable of producing a 28-in. mercury vacuum. The range of capacity is very large, the *Körting* machines dealing with from 10 to 20,000 c. ft. of air per minute.

HYDRAULIC MACHINERY

BY

ALFRED TOWLER, M.I.Mech.E.

Hydraulic Machinery

Introduction

Hydraulic machinery may be defined as machinery operated by water under pressure. Such machinery may be employed for a variety of purposes, such as, pumping, hauling, lifting, gun-training, and pressing; each in an infinite variety of forms for use in docks, boiler shops, structural steel-works, forges, armaments, baling, ply-wood and oil production. Generally the water pressure is produced artificially by automatically controlled pressure pumps. Sometimes a group of machines are supplied from a common system, in other cases each machine has its own pressure supply. The choice of the group or the separate system depends entirely upon the nature of the work to be performed. The group system is the one adopted by docks, industrial works, and hydraulic supply companies. Essentially it consists of a power pump, hydraulic accumulator, pressure, return, suction-pipes, and an overhead tank, together with valves and connections to the hydraulic machines which it has to serve.

In large installations having considerable mileage of service mains, in which it is desirable to maintain a fairly uniform pressure throughout the whole system, it is usual to provide several pumps and accumulators, either together or in separate supply stations. And, if such installations are liable to sudden local demands, provision must be made to meet the same; this is usually effected by locating the supply stations as near as practicable to the points of maximum consumption. It is also common, in these cases, to locate an accumulator at the extremities of the service main, these accumulators being loaded lighter than those at the pumping station and provided with fixed stops; they should preferably be fitted with automatic choking valves, so that they may approach their extreme full or empty positions slowly.

A ring range of pressure-pipes is one in which the extremities are joined, forming a ring with the supply, and all hydraulic machinery connected thereto would receive its supply from opposite directions, as this tends to equalize the pressure; but all such connections should preferably take the form of a " Y " rather than a " T ".

In laying out hydraulic systems, care should be taken to avoid air locks,

that is, pockets for holding air, and air cocks should be fitted at the higher points; if the locality is liable to frost, the pipes must be buried sufficiently deep below the surface of the ground. Alleviators or shock absorbers, sometimes called momentum valves, should be located wherever sudden changes of velocity take place, such as near machines using intermittently a relatively large quantity of pressure water; unnecessary bends, sharp corners, and high velocities should be avoided by the use of relatively large pipes, as the sudden stoppage of long columns of water at a high speed is a fruitful source of failure; furthermore, high velocity through tortuous passages tends to heat up the water, which damages leather and gutta-percha, and sometimes causes valve seats to come loose, and thence a serious breakdown.

The pumping main, that is, the pipe connecting the pump with the accumulator, should be separate from the service or delivery main to the machines. Each pumping unit, whether it be single-, two-, or three-throw, double or quadruple action, should have a separate suction supplied with water under an effective head of not less than 10 ft. The use of an air or vacuum vessel supplied with air under pressure is a great advantage to large pumps which run intermittently, as it forms an elastic cushion when the flow is stopped, and a reservoir to draw from when the pumps start; this accessory is of particular advantage where the pumps start and stop very rapidly.

There are two methods in common use for obtaining pressure on the suction: the open circuit, in which water is drawn from an open-top overhead tank, into which the return water from the hydraulic machines is discharged; and the closed circuit, in which water is drawn from a closed vessel, into which the return water is led against an air pressure maintained artificially by an air compressor or from a compressed air supply; in the latter the suction head may be easily altered if desirable.

Water most suitable for hydraulic systems is rain or distilled water free from acid or grit; hard water should never be employed, as it not only increases friction in glands and leathers, but forms internal encrustation in the pipes and passages. To maintain the water free from grit, strainers mostly of copper gauze are employed, and these are more accessible in the open circuit when located in the overhead tank.

The periodical addition of soluble lubricant also free from acid reduces friction and adds to the life of leathers and packings, keeping the bearing surfaces in better condition. Frequently in oil presses oil is used as the pressure fluid, having the advantage of not being affected to any material extent by frost. Usually the same oil is used in circulation as is produced by the presses, in order that no damage shall be done to the product should there be a leakage in the oil under pressure.

In the lay-out of a hydraulic installation the pumping station should be located in a central position. Where the machines to be operated are in close proximity only short main is needed, and a higher velocity of flow can be reckoned on; if, however, the machines are fixed at considerable

distances from the power station and each other, a slower velocity in the supply main is necessary in order to avoid material loss in pressure, and shock arising from sudden change or velocity.

When the demand for hydraulic pressure is fairly constant a single pumping unit may be sufficient, but, when there is considerable variation in the demand, multiple pumping units are preferable, as these, when automatically controlled, can be readily set to cut in and out of action according to the requirements of the machines. Automatic control gears are generally operated by the accumulator, and those electrically operated are usually quicker in action than the mechanical ones, tending to cause surges in pressure, and necessitating the adoption of cushioning devices, such as alleviators, &c., already referred to.

When additional hydraulic machinery is installed remote from the existing central station, it is sometimes found that the mains are too small to carry the extra power required, and it may be found cheaper to install an additional pumping unit than incur the cost of new main. In fixing the size of main, present and future requirements, and also internal and external corrosion must be considered. Solid drawn-steel mains are the most reliable for heavy pressure and ground which is liable to settle, but they are more susceptible to corrosion than cast iron, and they should on no account be laid in ashes or earth containing salts which attack steel or iron.

Certain kinds of work require two pressures, the higher pressure being employed for the final squeeze and usually obtained from an intensifier; but when there are a number of machines requiring such a pressure a separate high-pressure service is installed, having its own pumping unit and accumulator.

In oil extraction and similar work where the process must of necessity be slow, whether the material is pressed in canvas-covered layers between multiple plates or plattens, or in a cylinder or box, perforated in order to allow the fluid to escape, a power pump without accumulator may supply a group of presses.

Direct power pumps are often made with rams of different sizes arranged for two or more pressures, and when working on the low pressure all the rams are in action, the larger rams being automatically cut out when a predetermined pressure has been reached. Such pumps are economical in power, and well suited for large baling presses, where the pressure required is small at the commencement and rises rapidly at the completion of the operation.

Leather and gutta-percha jointing material for cold water and pressures up to 1 ton per square inch are excellent, provided the rings are let into recesses so that they cannot be blown or squeezed out, the commonest form of recess being the spigot and socket, and, in such cases, the meeting surfaces which grip the jointing material should be grooved in order to increase the grip.

Leather and sometimes gutta-percha rings are made in one piece cut from a sheet, but more commonly the latter is made from round or square

cord, cut to length, and the ends heated and held together until they adhere, on account of the saving in material.

For higher pressures than 1 ton per square inch soft copper grooved rings answer well; and in small pipes of solid-drawn steel or copper the joint may be made metal to metal, in which case the end of one pipe is machined square and the other V-shaped, so that the narrow edge is forced against the flat surface; for metal joints the flanges and bolts require to be extra heavy.

Various pressures have been adopted for different purposes at different times, hence the great lack of uniformity which now exists; 750 lb. pressure per square inch is general for dock work, and 1500 lb. pressure per square inch for industrial works; the working pressure in docks, iron and steel works may vary from 500 lb. to 1000 lb. per square inch, the old installations having adopted the lower pressure; in structural works from half a ton to 1 ton pressure per square inch in different districts; while some ordnance works have adopted a higher pressure still. The disadvantage of a low pressure being in the size of the pipes, cylinders, rams, &c., and that of the high pressure in the increased difficulty of keeping valves, glands, and joints tight; the present tendency is to adopt a uniform medium pressure for all purposes, namely, 1500 lb. per square inch.

CHAPTER I

Valves

Valves may be divided into four classes:

1. Stop valves.
2. Self-acting valves.
3. Hand operating valves.
4. Automatically operating valves.

Class 1.—Stop valves are mostly of the screw-down type, with either inside screws or outside screws, the former being the more common. The valve, seat, and spindle are generally of bronze, and the valve and seat faces (beats) are mostly "mitre" (45°). Smaller valve bodies are usually of bronze, in which case the seat is generally in one with the body. Medium and large valve bodies may be of cast iron, cast steel, or forged steel, according to the size and pressure. Valves which, owing to the size or pressure, would be difficult to operate are either balanced or the pressure relieved by external or internal by-passes, the size of valve and working pressure determining when the change must be made. Where oil under pressure is used, the stop valves and distribution valves are sometimes case-hardened steel balls, in which case the body is usually of forged steel without seats. Valve bodies are usually of the straight-through type, although angle bodies

are sometimes used, particularly for high pressures in which the bodies are of forged steel.

Class 2.—Self-acting valves—that is, valves which are operated by the change in flow, such as, pump valves, check or non-return valves, the most common form being the " mushroom " or " wing " valve with mitre faces—when well guided above and below, are easy to make and keep in order, and are durable and efficient when the diameter and lift are relatively small. These factors are determined by the volume of fluid delivered and the number of operations the valve is required to make in a given time, and, in the case of pump suction valves, whether the water is supplied under a head or has to be lifted, the former supply being essential for high-speed working.

High-speed pump valves should never exceed 5 in. in bore of the seat, even with overhead water, nor should the lift exceed $\frac{5}{16}$ in., and smaller valves should have a proportionately smaller lift. If a greater area is required for the volume of water to be dealt with, either clusters of small valves or annular valves should be employed; and in the latter case " flat " faces are easier to make and keep tight, but they cause a little more disturbance to the fluid passing through than mitre faces on account of the increased deflection of the current. The valve lift should be positively stopped, and rubber cushioning rings or springs under compression should be used to accelerate the closing of the valves, which, to be durable and efficient, must close before the return flow takes place, otherwise they damage their beats. Case-hardened steel balls may be used satisfactorily with oil under pressure, but are useless with water on account of being liable to pit.

All valve (including stop-valve) seats, particularly for the higher working pressure, should be secured, otherwise they are liable to work loose, as the water becomes warm when flowing through tortuous passages at high velocities; and the coefficient of expansion of the material of which the seat is made is often so much greater than that of the material of the body which surrounds it that the seat is permanently crushed. A loose suction seat stops the flow and nothing further happens, while a loose delivery seat is often the cause of a serious breakdown. When there is a joint in the valve body at the seat, the latter can be positively enclosed, but when the body is in one the seat must be secured (*a*) by being expanded into a groove or grooves, (*b*) by riveting, (*c*) by screwing, (*d*) by being pinned, (*e*) by being held down by a cage.

Class 3.—The most popular hand-operated valve for hydraulic machine tools and cranes, when the pressure does not exceed 1 ton per square inch, is the balanced leather-packed piston valve. It will remain " bottle " tight longer than other types of valves, and generally give warning before the leathers, which may easily be renewed, fail. Small slide valves are sometimes employed for pressures not exceeding 1000 lb. per square inch, on account of their simplicity, but they become too hard to work by hand as the size and pressure increase, and the faces are apt to score. For high

pressures, balanced mitre valves are employed, the small sizes being operated direct, and larger sizes hydraulically operated by a pilot valve. The favourite control valve for baling presses is the screw-down type operated by hand wheel or swape handle. The control valves of the shell presses adopted by the Ministry of Munitions were of the lever-operated double-threaded screw-down type, the pressure valve being balanced and the pressure and exhaust valves being operated by separate levers.

Class 4.—Automatically operated valves, that is, the working valves of a hydraulic engine. In this class it will be readily understood that the conditions of continuous work are severer than in any of the classes of valves already considered, and for this purpose some form of slide valve is commonly used, which in its simplest form is flat, but may take the cylindrical form as in the "trunnion" valve or the "eccentric" and "disc" shape. Further, in order to reduce the friction in case of high pressures, balancing devices are sometimes fitted, and balanced piston and beat valves are also used, and may be used for higher pressures than the plain slide valve.

CHAPTER II

Accumulators

The accumulator is a mechanism for storing energy, determining the pressure, and keeping the same constant in a hydraulic system. It consists of a loaded ram located in a cylinder provided with a gland, a connection to the pressure main, and a base for securing on foundations. When weight loaded, the ram is vertical, when pressure loaded, it may be either vertical or horizontal.

In the best form of weight-loaded accumulator the cylinder is fixed with gland above, the ram moves and is attached to the weight by cross-head and suspension rods; in the cheaper form the ram is fixed, while the cylinder floats and is attached to the weights, with the gland inverted. The weight may be built up of concrete or cast-iron segments, resting upon a table secured to the tension rods or cylinder as the case may be. When loose loading material is employed, such as scrap, punchings, or ballast, a wrought-steel casing is used. With a fixed cylinder the casing is provided with an internal tube sufficiently large to clear the cylinder, the loading material being packed in the annular space, and the tension rods attached either to the tube or the bottom of the casing. In the floating-cylinder type no inner tube is necessary, the casing bottom or table being secured to the cylinder.

Weight-loaded accumulators may be either of the self-guided or the externally guided types (fig. 1). The fixed cylinder lends itself best to the former, two or four machined guide ribs being formed on it, and runners

or shoes fixed on the casing or table to guide the load and prevent it from turning. This type is suitable for small accumulators, and also answers quite well for large ones, provided their height is limited and the locality where they are finally fixed sheltered, as the leverage is considerable when the accumulator is at its highest position, so that it may be dangerous if subjected to high wind pressure. The externally guided type has no limit to height except that of manufacture; for it will be understood that the

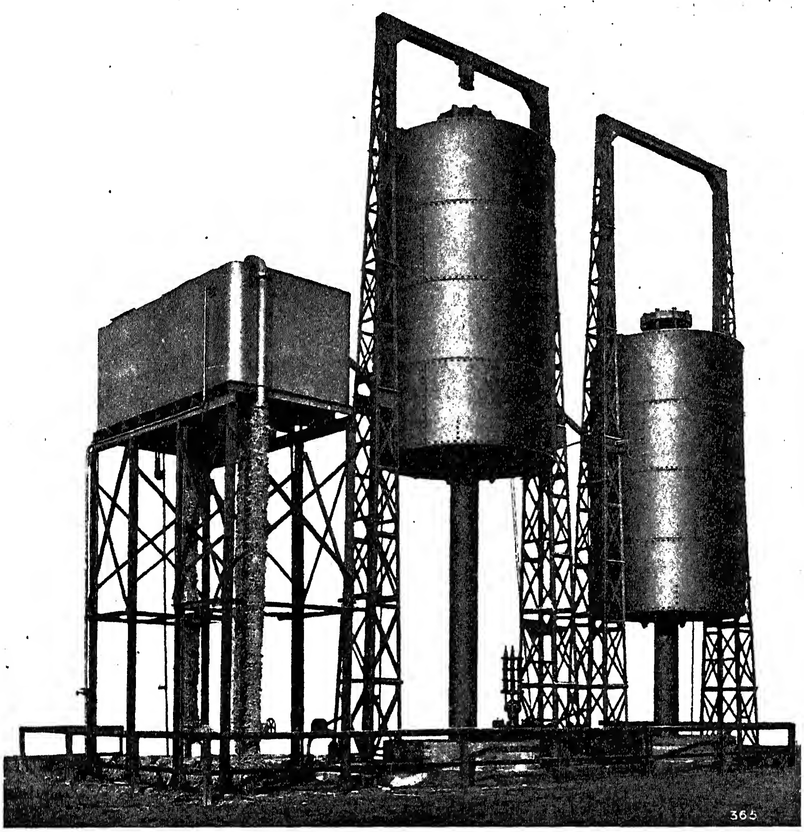


Fig. 1.—Externally-guided Hydraulic Accumulator

same loading for a given diameter of ram produces a given pressure irrespective of the length of the stroke, consequently, the longer the stroke the greater the capacity, and hence the advantage of a long stroke from that point of view. On the other hand, the risk of serious damage is greater with a long stroke should the pressure fail and the accumulator descend unchecked, and provision should be made for this event by inserting timber banging blocks on which the load can strike and rest when the accumulator is down. In order to lessen the blow, large accumulators are often provided with a choking valve, which gradually closes the outlet

as it approaches the banging blocks. Fixed stops are sometimes used, particularly for the light accumulators, to prevent the ram from being forced out of the cylinder when it is not desirable to employ other restricting devices, and various methods are employed for doing this, such as collar or bayonet end formed on the ram, arranged to catch the neck of the cylinder, or a projection on the load casing arranged to catch a similar projection on the outside of the cylinder, or the loading is held by anchor chains attached to the foundations, or spring buffers fixed on the guide structure.

Guide structures may be built up of timber or rolled-steel sections, stayed, strutted, and secured to the foundations; shoes or runners being fixed on the loading, which slide on rails or rolled-steel sections; it is customary to fit small positively stopped accumulators with relief valves, and large ones with automatic double-acting choking valves.

When accumulators have no positive stops, an automatic deflecting valve is fitted, which, in its simplest form, is opened by positive mechanical means, when the loading has reached a predetermined height, and allows the delivery from the pump to be deflected into the return main to the supply tank, a check valve being combined with this valve to prevent the water in the accumulator from returning, and closed immediately the loading descends sufficiently, thus loading the pump again. The disadvantage of this arrangement is that the pressure on the pump is only partly relieved when the deflecting valve is used to govern the supply, and the accumulator ram becomes worn between the points where the deflecting valve is opened and closed, which is relatively a small fraction of the accumulator's stroke; therefore, this type of valve should only be used for small accumulators, or as a relief valve to prevent the accumulator overrunning its stroke.

Hydraulically operated deflecting valves can be set to open fully at the highest position of the loading, and close at any predetermined position of its stroke, and when a deflecting valve is required to both govern the supply and determine the stroke, this type is preferable. As an additional safety device the end of the ram may be drilled, so that when it passes the gland packing it allows the pressure to escape.

Pressure-loaded accumulators are mostly used afloat, and are also specially applicable where the pressure or velocity is subject to considerable and continuous change, to equalize and govern the supply rather than to store energy. The pressure media is either steam or compressed air, the disadvantage of the former being heat and condensation, and on account of its comparatively low inertia this type of accumulator can operate at considerably higher speeds than would be considered safe in a weight-loaded type, and an accumulator of smaller capacity may be employed.

CHAPTER III

Intensifiers

A hydraulic intensifier is a mechanism for increasing the pressure. In many kinds of press work the normal hydraulic pressure is sufficient for all purposes except the final squeeze, which may require from two to four times the pressure for a relatively small travel.

The simplest form of intensifier is the one which is charged simultaneously with the press or presses it serves. If an intensifier supplies more than one press requiring high-pressure water at different times, it is necessary to insert a high-pressure control valve on each press. Intensifiers of this kind are usually vertical, and, when the ratio of the pressures will allow, are made of the telescopic type, that is, the low-pressure ram becomes the cylinder for the high-pressure ram in order to shorten the height. Such an arrangement is shown in fig. 2. This device consists of a low-pressure cylinder fixed upon its foundations, carrying a crosshead on columns. Connected to the crosshead is the high-pressure ram, while the low-pressure ram, which is the only moving part, slides up same and serves as its cylinder. Two glands, a control valve, a check valve, and pipe connections are all the fittings that are necessary. Constant low pressure is admitted to the high-pressure ram, which is hollow, a check valve being provided to prevent the intensified pressure from escaping into the low-pressure main, and the control valve is fixed either on the low-pressure cylinder or near the press which the intensifier serves.

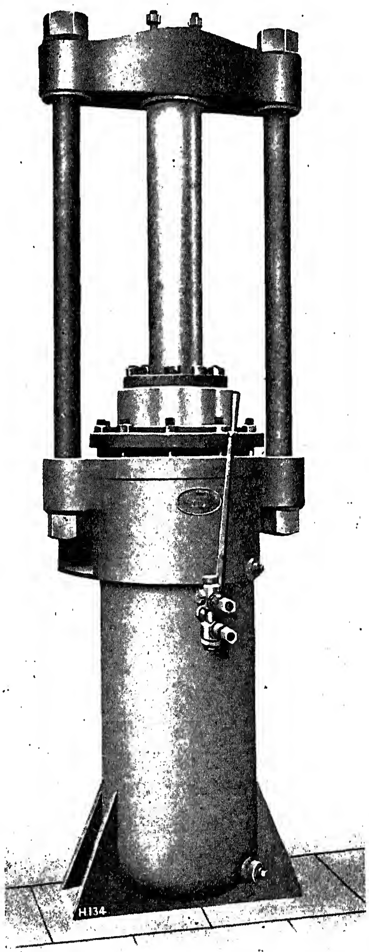


Fig. 2.—Intensifier

The intensifier being initially fully charged, the moving cylinder having been driven into its lowest position by the constant pressure, the method of working is as follows: when the control valve is opened the pressure is admitted to the stationary cylinder

and forces the moving cylinder upwards; the water contained in the latter is forced through a hole in the smaller ram past the check valve, which prevents it returning to the low-pressure main, into the press at a pressure intensified in inverse proportion to the areas of the respective rams. Then the return stroke is made by reversing the control valve, thus opening the stationary cylinder to exhaust; the low-pressure water will then open the check valve and fill the moving cylinder, forcing it into its lowest position in readiness for the next intensified operation. When the ratio between the accumulator pressure and intensified pressure is small, the telescopic principle cannot be employed, and the two rams require separate cylinders.

When a number of presses require intensified pressure, two separate pressure systems may be adopted, the high-pressure service being obtained either by a separate pump and accumulator or from a double-acting automatic intensifier, which is really a double-acting pump using the low-pressure power to " boost " up the pressure, the advantage being a continuous supply in a separate main, which may be admitted to any press as required. Accumulators are not required with intensifiers of this kind, as the mechanism stops, starts, and varies its speed according to the demand.

It will be understood that both the intensifier and the two pressure systems are water-saving devices, but are distinct from the apparatus for which the term has a specific meaning; that is to economize pressure water when the machine is being operated without doing work, the essence of water-saving devices being the provision of means for operating a hydraulic machine by a smaller auxiliary power, and only employing the maximum power when it is actually needed.

Water saving in some form can be applied to nearly all kinds of hydraulic machinery, but whether it is worth doing depends upon the percentage of saving, this varying with the type of machine and work to be done. For example, the tappet gear on the operating valve of a hydraulic manhole punch can be so set that there need be no superfluous stroke. The possible percentage of water saving would be negligible, save, perhaps, when there is a frequent change of tools; while in a flanging and forging press operating on deep work, the working stroke may be double the stroke on which full power is required; and, furthermore, a good deal of manipulation is necessary in such machines, when the dies or work are changed, to see that the requisite clearance has been allowed for before the work is put in, and consequently in this case the percentage of saving may be considerable.

There are two types of water-saving device, namely, the voluntary and the automatic, the voluntary type being also subdivided into two classes, (*a*) the single-lever operated type, and (*b*) the independent double-lever operated type.

In type (*a*) the control valve for the water-saving ram or rams is operated by the same hand lever which operates the valve for the main ram but in advance of the latter, the hand lever being provided with a notched sector. The operator can manipulate the moving table, the whole or part of its stroke, or up to its work, on the lower power, by moving the control lever

from the neutral notch to and fro in the exhaust direction or in the power direction up to the first notch, and during this power stroke the main ram draws in overhead water through an auxiliary check valve. When it is desired to employ the full power the operator moves the control lever from the first to the second notch, and the auxiliary check valve is closed by the inrush of pressure water.

Type (*b*) works in the same cycle, but in this case the operator must remember to move the separate control levers in sequence. In the automatic water-saving device, the attendant operates the control valve of the water-saving ram or rams only, and when they are overpowered the rise in pressure operates the control valve of the main ram in water-saving device which is of the hydraulically operated type. Thus water saving is made compulsory.

As there is a danger of the larger rams intensifying the smaller rams by defective or improperly worked valves, relief valves are always fitted, preferably of the pressure-loaded check-valve type, which allows what would otherwise be intensified pressure to escape into the pressure main.

There is a general belief that water-saving devices slow down hydraulic machines, and that when their use is optional such devices are not used to the fullest extent; but in modern practice this slowing down has been removed by increasing the size of the auxiliary check valve, and operating the same by a small hydraulic ram connected to the supply to the water-saving ram or rams from a common control valve, this enabling the auxiliary check valve to be opened on the exhaust stroke as well as the water-filling stroke. To accelerate the closing of the auxiliary check valve it is spring loaded.

CHAPTER IV

Riveters

Whenever the highest quality of riveting is essential, a hydraulic riveter is employed, for it adjusts itself automatically to the varying thicknesses of work and applies a definite pressure to each rivet, and, moreover, when operated intelligently, the power is held on until the rivet has cooled sufficiently to ensure the meeting surfaces of the work riveted being securely held close together, and the rivet filling the hole like a fitted bolt, little or no caulking being necessary.

Soft-iron rivets up to $\frac{3}{8}$ in. diameter may be riveted hydraulically cold, but a section of such work reveals its defects and the necessity of it being closed hot. Rivets up to $\frac{1}{2}$ in. diameter are sometimes put in place cold and heated up in the operation of being hydraulically closed by passing a current of electricity through the snaps; but the general practice is to heat the rivets in a suitable furnace before they are put in place to be riveted.

Work which is subject to pressure, such as boiler work, needs d the closing power that is sufficient for structural work, and when the j exceed $1\frac{3}{8}$ in. in thickness it is desirable to fit the riveter with a plate-cl device, which squeezes the plates together prior to and during the oper of closing the rivet. This takes the buckle out of the plates and pre the rivet from being forced between them in the form of a " flash ", v is detrimental to finished products.

Heavy-powered riveters are often made with two or more powe order that they may be usc lighter work. A good com tion is two powers, the low which may be used as dc for either lighter work or closing in combination witl higher power. The addit powers are usually obtainc combination with a seconc linder, either located in the ram or main cylinder cover control valve being commc all the powers which are obta by operating a stop valv valves. For example, two pc may be obtained by shuttin the small cylinder for the power and employing botl the larger; three powers shutting off the large cylin shutting off the small cylin and employing both cylin Some makers obtain the se or higher power by emplc a small intensifier, located ir main cylinder cover, of su capacity as to allow for squeeze and the stretch of the frame. Water-saving devices are also f when desired.

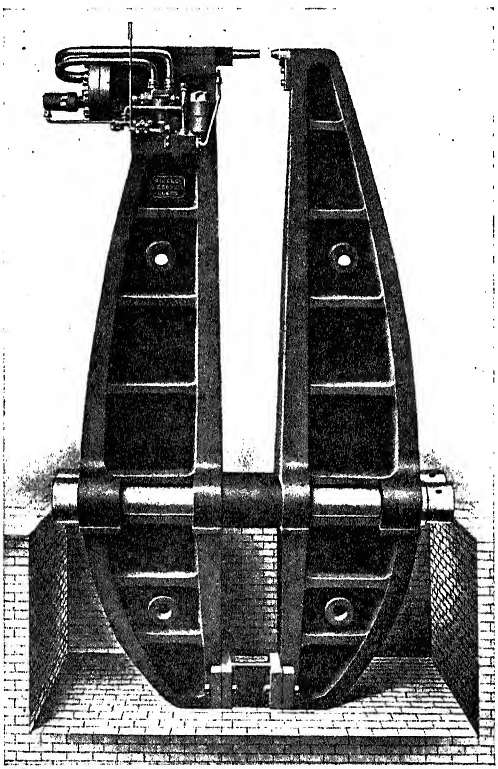


Fig. 3.—Fixed Built-up Hydraulic Riveter

There are two forms of riveters in common use, namely, statio and portable. A stationary riveter, as the name implies, is one whic fixed upon a foundation, and is employed upon work which is suspe in a crane during the operation of riveting; such work as the shell firebox of Scotch and locomotive boilers, the shell of Lancashire Cornish boilers. The opening is the distance between the riveter a while the gap is the depth of the opening from the centre of the sr The gap is made to suit the class of work, and heavy riveters with gaps are usually built up of cast-steel arms on a cast-iron distance p

secured with massive forged-steel bolts, which are heated before being finally screwed up. The head is also bolted on, and should preferably be of the flush-top type, fitted with adjustable slides to take up the wear, as shown in fig. 3. The gap of some locomotive riveters often exceeds 20 ft., and for marine boilers the gap is usually less than this but the power greater. Whenever practicable the riveter body is made in one piece of cast steel,

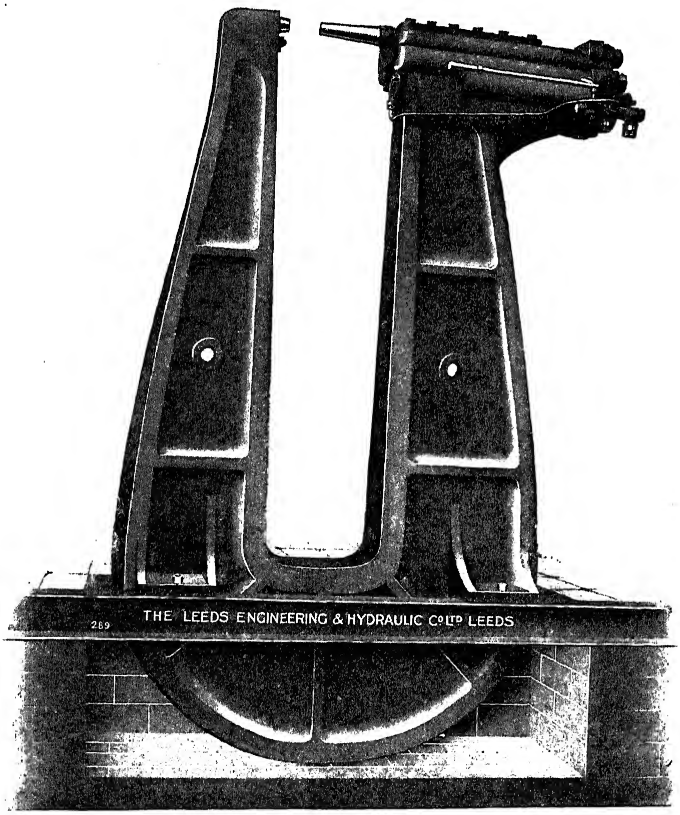


Fig. 4.—Fixed Bear-type Hydraulic Riveter

the head being bolted on one arm, and the holder-up snap on the other (fig. 4).

For tube or small barrel work, the holder-up arm, or hob, is usually made of high-tensile forged steel. Sometimes the plate closer is fixed on the holder-up arm, but more preferably to the cylinder head in view of the operator. In the highest quality of riveted work it is essential that the power shall remain on the rivet after it has been closed, so as to allow it to cool sufficiently, the duration of the dwell being determined by the size of the rivets; for, when riveting is done hurriedly, the duration of the dwell may be curtailed to the detriment of the work, and the amount of

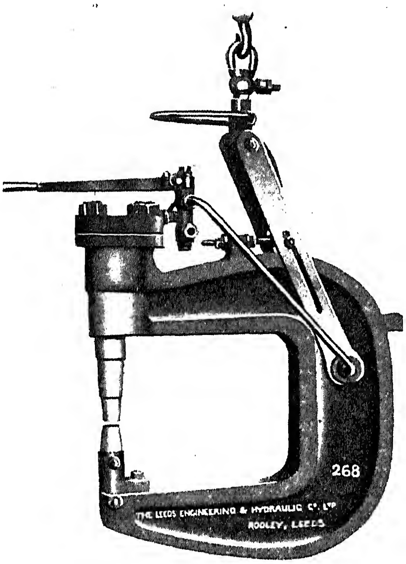


Fig. 5.—Portable Bear-type Hydraulic Riveter

caulking to be done on any piece of work, in order to make same tight under pressure, is a safe indication of the quality of the riveting.

To ensure a uniform dwell, timing devices are sometimes fitted, in which the adjustable timing mechanism is either locked up or under seal, the common principle of which is to convert the three stages of operation, namely, closing, dwell, opening, into one complete cycle in which the operator only starts the cycle; but, if from any cause the operation must be stopped before it is completed, there is often a difficulty which may be serious.

There are three distinct types of portable hydraulic riveters in common use, namely, the "bear", the "hinged", and the "link" type. In the bear type (fig. 5), which is the lightest, cheapest, and simplest, the cylinder and frame are in one. The hinged type, or sometimes called "scissors" type (fig. 6), and the link type can be used in positions which are inaccessible to the bear type, as the cylinder is located at the opposite end of the arms to the snaps. These are mostly used on structural work. It is essential in good work that the snaps shall keep in alignment, and in the bear type the only working parts are the piston and ram, which are usually eccentric with each other. They will withstand considerable wear, and when worn can be rebushed inexpensively. The hinged type will keep in alignment considerably longer than the link type on account of its stiffer construction and superior guiding and bearing surfaces.

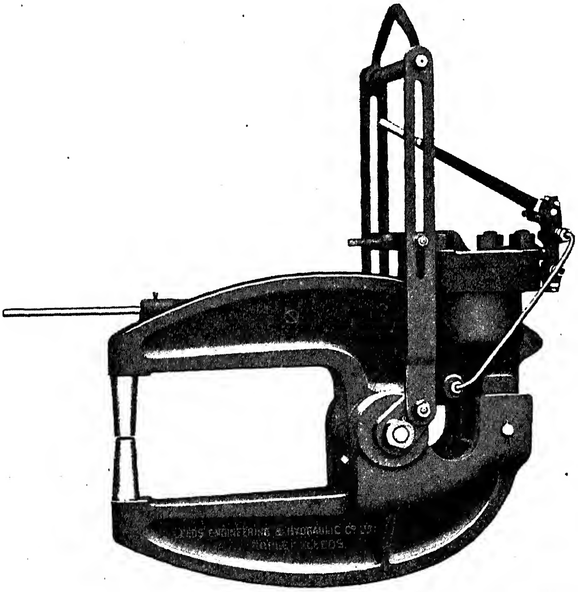


Fig. 6.—Portable Hinged-type Hydraulic Riveter

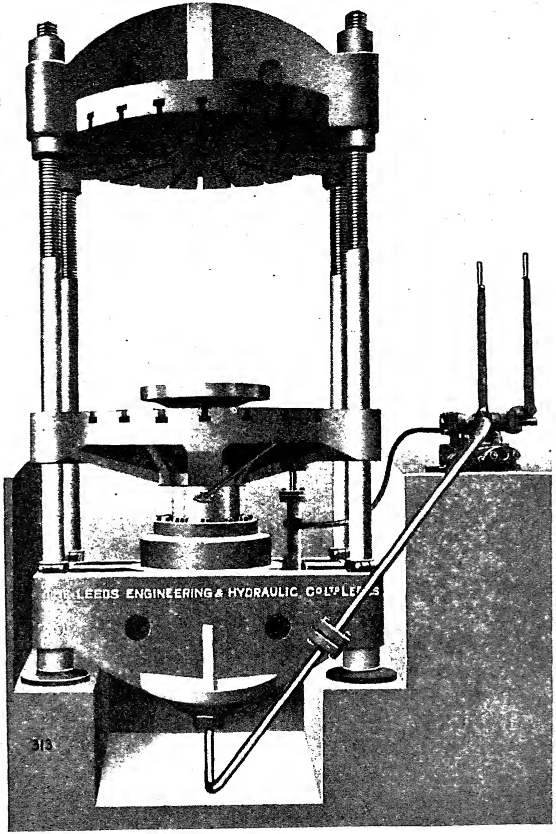


Fig. 7.—Four-column Hydraulic Flanging Press

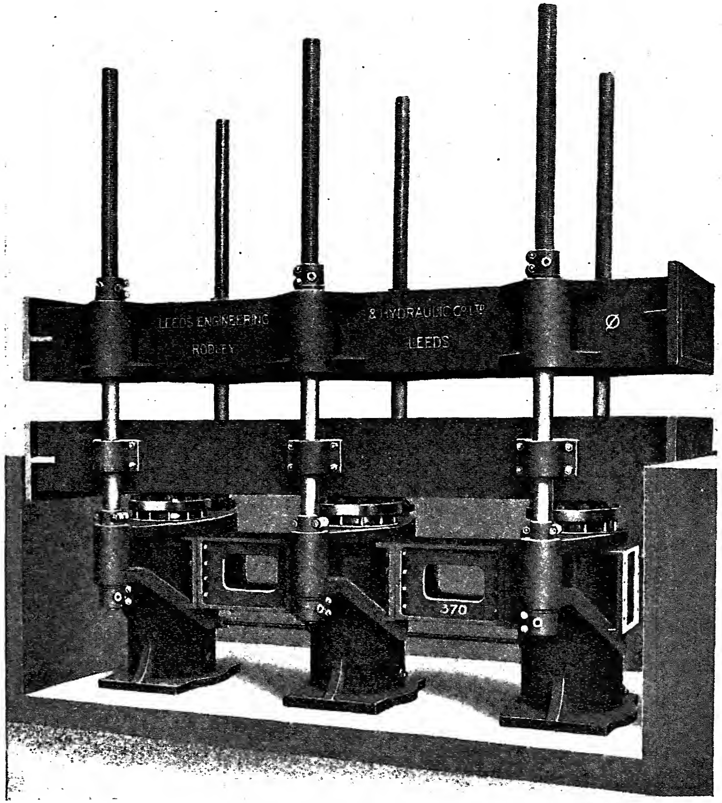


Fig. 8.—Hydraulic Frame Press

A desirable factor in portable riveters is handiness, and in this weight plays an important part; and although the material may be, and often is, highly stressed, the weight is really determined by the power, gap, and opening. These in turn are determined by the work to be done, such as the diameter of the rivet, the distance of same from the edge of the plate, and if there are any flanges on the edge of the plate, as the riveter body and arms must clear same.

Portable riveters are usually suspended from the monkey carriage of a jib crane, either by chain blocks or hydraulic jack lift, but the attachment to the riveter itself is of supreme importance, and depends upon the nature of the work. There are four forms in general use: (*a*) the eyebolt or shackle suspension, (*b*) plate hanger, (*c*) bow hanger, (*d*) compound hanger. Each method of suspension is equally applicable to the hinged type. When the riveters are heavy, the bow and compound hangers are fitted with worm and wheel gearing. Portable riveters are sometimes fitted with two powers for light and heavy work.

From what has been remarked upon the type, power, gap, opening, and method of suspension, it will be gathered that, to obtain the best results in the shortest time, the riveter must suit the work; and, if the work varies materially, an assortment of riveters should be available for employment on the various classes of work.

CHAPTER V

Flanging Presses and Manhole Punches

FLANGING PRESSES

In Lancashire, Cornish, and Locomotive boilers the end plates and fireboxes, having been heated uniformly all over, are flanged and dished in a four-column hydraulic press, such as shown in fig. 7, this press being usually equipped with four radial vice rams which project through slot holes in the moving table, the cylinders resting on machined seatings forming part of the base, and secured by tee-headed bolts, so that they may be fixed at the required distance from the centre of the press to suit any particular work.

These rams are operated by a hand-control valve fixed alongside the main control valve, and are used to hold the plate in position against the die secured to the head. The centre ram, which is located inside the main ram, is useful for flanging holes and doing smaller work, and an auxiliary ram located in the head if provided is useful to facilitate stripping. The head is made adjustable to suit the different depths of dies and work.

Fig. 8 is a type of hydraulic frame press usually employed upon bridge flooring, girder, and such like work; the three cylinders are located below, and can be operated separately or together, giving three powers or six

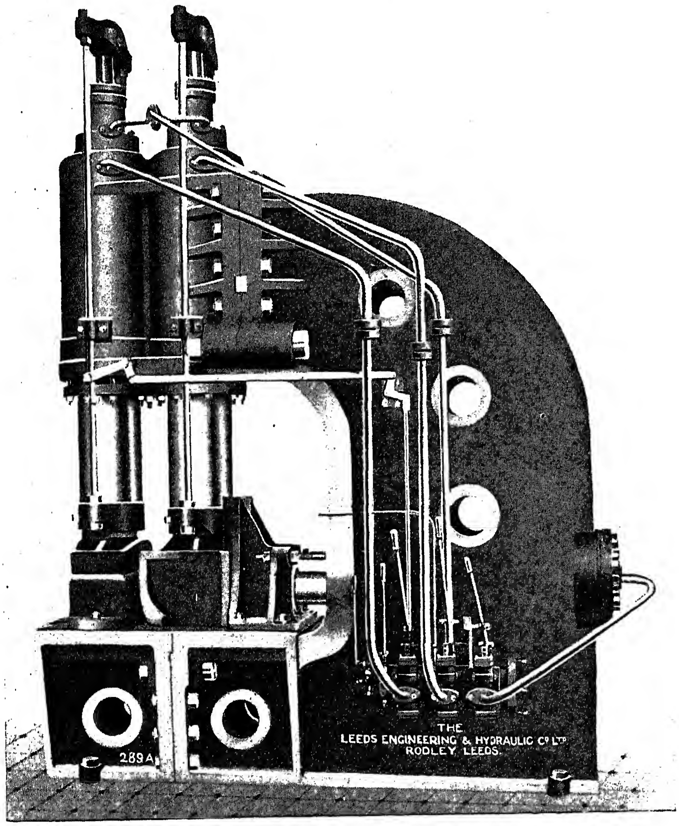


Fig. 9.—Progressive-type Hydraulic Flanging Press

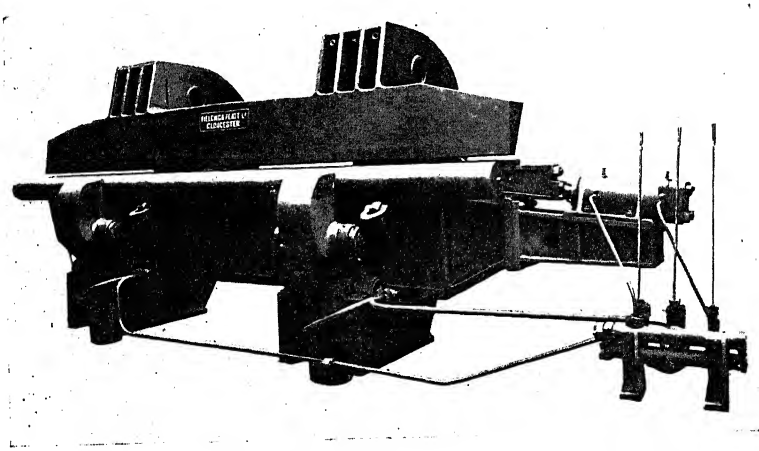


Fig. 10.—Hydraulic Garboard Bender

powers when worked in conjunction with an intensifier. The head is adjustable to suit dies for deep and shallow work.

Fig. 9 shows a form of flanging press known as the open progressive type, and is used largely for flanging end plates and furnace mouths of marine boilers. The outer vertical ram holds the plate on the die, while

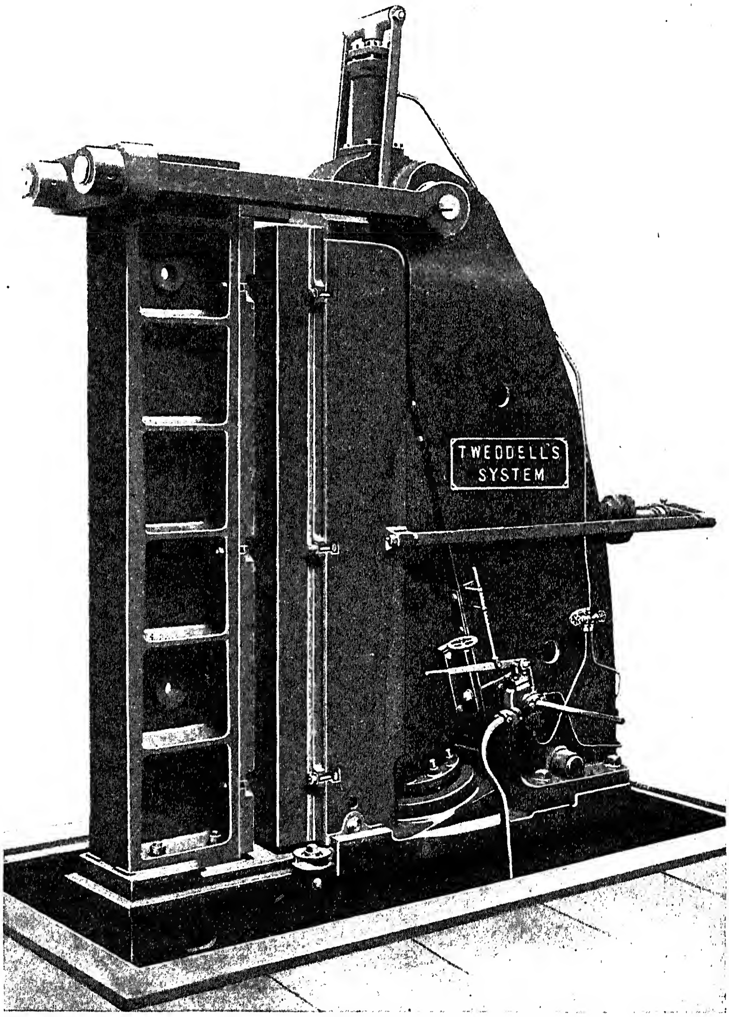


Fig. 11.—Vertical Hydraulic Plate-bending Press

the head of the inner ram bends the part to be flanged over. Finally the horizontal ram head finishes it up to the required shape, part of the edge of the plate being heated and flanged; then an adjoining part is treated in a similar manner, until the whole is completely flanged " step by step ". Sometimes the two vertical rams are coupled together to do single operations needing greater power.

Fig. 10 shows a hydraulic garboard bender, which is used for bending steel plates cold for ships' keels and for doing all kinds of straight flanging for floor plates, bulkheads, intercostals, tanks, and other ship work. The angle assumed by the plate at one end may differ from that at the other end to any extent; that is to say that, while the plate may remain flat at one end, its other end may have both sides bent to any desired angle, the bending being performed in two operations, one side at a time. The plate is firmly gripped between a pair of girders by a powerful wedge action, operated hydraulically; the bending and flanging of the plate is then carried out by forged-steel rollers carried in a pair of cradles all operated by hydraulic cylinders and rams. These cylinders can be operated independently, so that any desired angle may be obtained on the plate. The roller is adjustable for bending various thicknesses of plates.

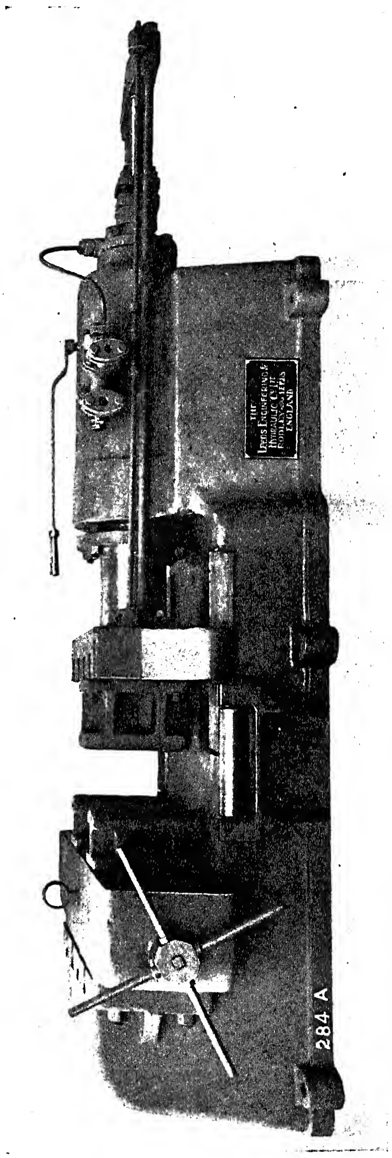


Fig. 12.—Horizontal Hydraulic Bending Press

Fig. 11 shows a hydraulic boiler shell-plate bender, which is simpler, quicker, and less costly than the ordinary power-bending rolls, and has the following additional advantages:

1. Plates can be bent to a true curve right up to the edge without leaving a flat.
2. Narrow curved plates can be also bent easily and accurately.
3. It is impossible to overload the machine.
4. It occupies very little floor space, and its upkeep is small owing to the few moving parts.

The bending girders are fitted with loose dies, so that special dies may be fitted when required for bending sharp corners and similar work. The bending girders and main frame are mounted and secured on a base containing the main cylinder, which operates the moving girder by a wedge action hydraulically operated, the pullback cylinder being fixed at the back of the frame, and its ram connected to the moving girder by a crosshead and side rods. The top of

the fixed bending girder is secured by a hinged clip, which is hydraulically opened in order to remove the plate when it has been bent into a circle. The cold plate to be bent may be fed into the machine in either direction edgewise, variable rates of feed being provided for feeding the plate through the dies, also adjustable stops and indicators showing the radius and thickness of plate; and the whole operation can be performed automatically.

Fig. 12 shows a simple type of horizontal hydraulic press usually

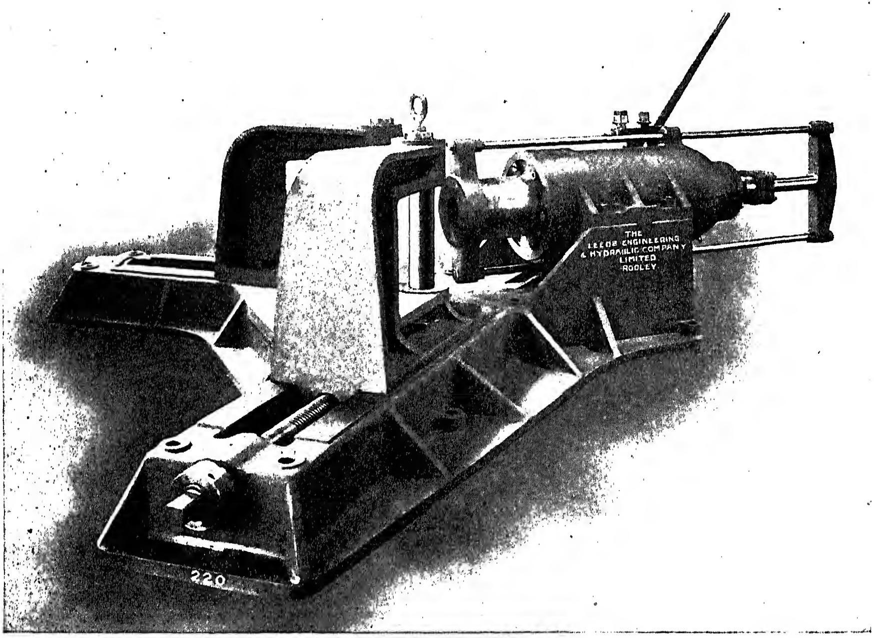


Fig. 13.—Horizontal Hydraulic Pipe Bend

employed in bending and straightening rolled-steel sections. It will be easily understood.

Fig. 13 shows a horizontal hydraulic press used for bending copper and steel tubes. The fulcrum pin brackets, which make an angle of 90° with each other, are mounted on slides formed in the bedplate provided with screws so that same may be set to the diameter and radius of the tubes to be bent; and a die is fitted into the ram socket and grooved rollers on the bracket pins. It is essential that the hollow in the die and groove in the rollers shall correspond with the outside diameter of the tube to be bent. Copper tubes are bent cold, and, when the angle of bend is considerable and short, they are filled with either sand or resin. Steel tubes are usually bent cold unless the radius is comparatively small, in which case they are bent hot.

Fig. 14 shows a vertical pipe bender fitted with telescopic hand pump,

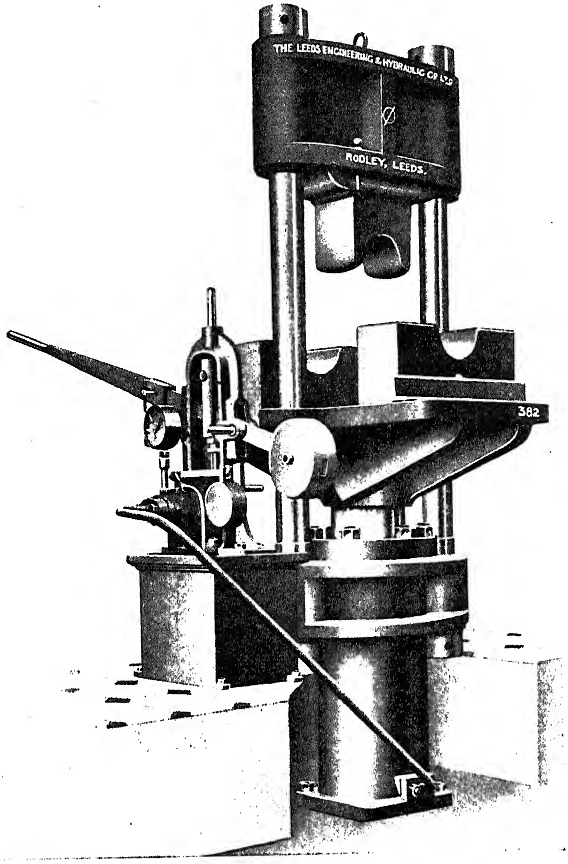


Fig. 14.—Vertical Hydraulic Pipe Bender and Hand Pump

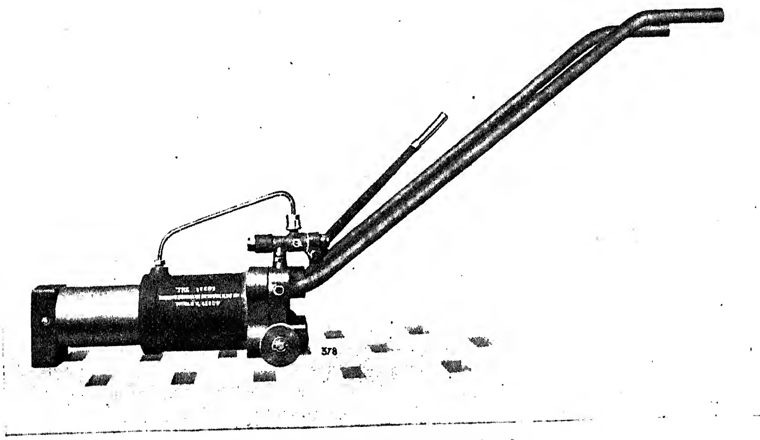


Fig. 15.—Portable Hydraulic Bender

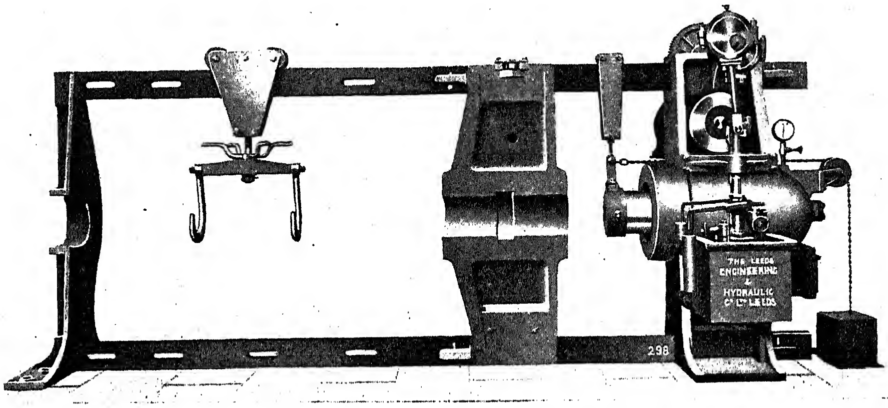


Fig. 16.—Hydraulic Wheel Press

the change-over from large ram to small ram being effected by a single-lever locking device.

Fig. 15 shows a type of portable hydraulic bender which is used very largely in shipyards for bending rolled-steel sections, such as girders,

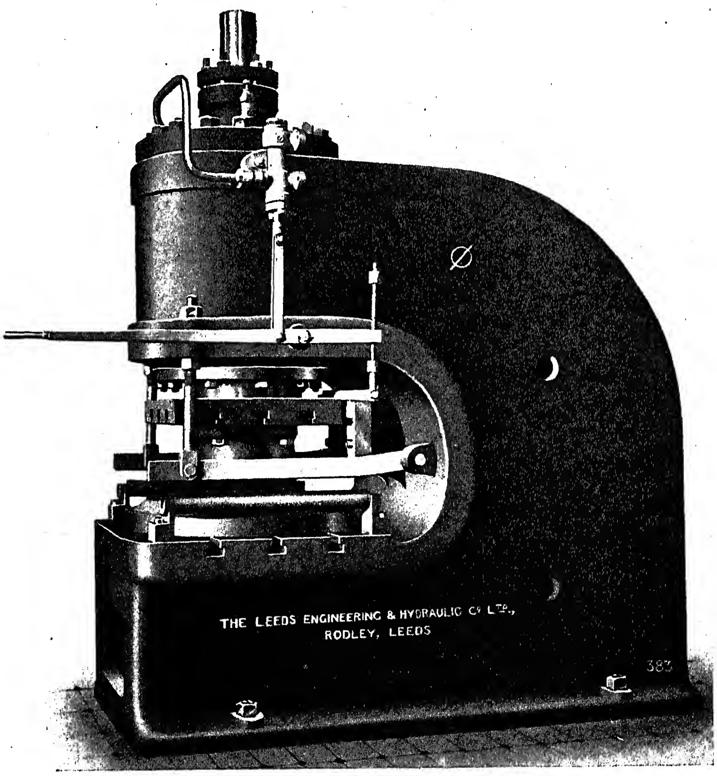


Fig. 17.—Hydraulic Manhole Punch

channels, bulbs, and angles. These are cramped on the plate floor, and are bent to the required shape by a portable bender, using fulcrum pin in the most convenient hole of the plate floor, the lighter sections being bent cold.

Fig. 16 shows a common type of wheel press used for forcing on and taking off locomotive, railway wagon, and car wheels, and is operated by an electrically driven two-throw two-pressure pump, in which the larger ram automatically knocks off, and the smaller ram relieves at predetermined pressures, the main ram being drawn back by weight. The wheels are usually rolled into the press and the axle lifted to the required height and suspended from the hooks of the adjustable monkey carriage.

MANHOLE PUNCHES

Fig. 17 shows a handy form of shipyard tool, the frame being of the open-sided type in which the punches and dies are arranged for fixing in two positions at right angles, so that oval holes may be punched either lengthway or crosswise; and the same machine with interchangeable tools may be used for bending, flanging, and joggling. The handiest joggling tools are those with adjustable jaws, which will take in a variety of work, such as angle, bulb, channel, and plate sections.

CHAPTER VI

Steam-hydraulic Presses

A very great expansion in the use of steam-hydraulic presses for forging and shearing has taken place during recent years. Formerly presses were mainly employed for the production of heavy forgings, their slow speed of working making them unsuitable for the medium and lighter classes of work; but the introduction of the steam-hydraulic press, and more particularly the improvements effected in its system of working, have entirely altered the position, and enable such presses to be employed in place of steam hammers with a considerable improvement in the output and appreciable saving of steam, as hammers are admittedly very wasteful in steam consumption.

The steam-hydraulic system of forging press operation is now so well known that a detailed description is hardly necessary. Speaking generally, the power is obtained by means of an intensifier, consisting of a steam cylinder working in conjunction with a hydraulic cylinder, the steam piston having a positive connection with the hydraulic ram. A steam pressure of say 150 lb. per square inch acting on the large area of the steam piston multiplies the pressure on the small area of the hydraulic ram, giving a

pressure in the hydraulic cylinder of the intensifier, and also, as they are directly connected, in the main cylinder of the press, of say $2\frac{1}{2}$ tons per square inch. A pre-filler or vessel containing water under an air pressure of say 60 lb. per square inch also forms part of the installation, its function being to keep the system full of water, so as to exclude air and to bring the press head down on to the work previous to the exertion of the intensified power.

Except in special cases, the steam-hydraulic press is now generally used for general forging work, as it is simple in working, capable of a much

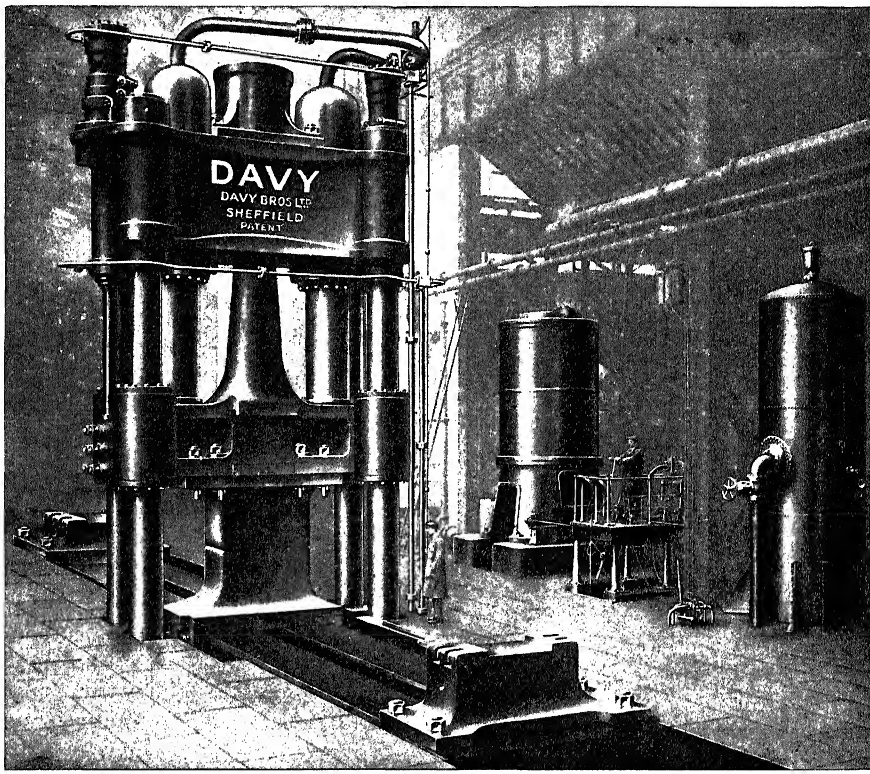


Fig. 18.—Steam-hydraulic Forging Press

faster working speed, and is more economical in power than any other system. Presses of this system are now installed in most of the principal steel works of the world, and they are made in powers varying from 100 to 12,000 tons.

The small presses of 100, 150, and 200 tons power are generally of the overhung self-contained pattern, in which the intensifier is attached to the back of the press frame, the latter being made hollow to form the filler. Larger presses are of the four-column type; usually the single-cylinder construction is employed for sizes from 300 to 2000 tons. Presses of over 2000 tons power are made of the special duplex-cylinder construction, as shown in fig. 18, representing a 4000-ton press.

The features of the duplex construction are the provision of two main cylinders instead of one, the absence of rigid connections between the rams and crosshead, and the provision of a central guide cylinder for the crosshead stalk, which have the effect of enabling the presses to bear without injury the heavy side strains set up when the work being forged is not directly under the centre of the press—a condition which is unavoidable in some classes of work and a serious factor in presses of large sizes.

A very important feature is the patent single-lever automatic controlling gear by means of which all the movements of the press are controlled from one handing lever. The gear is so arranged that the movement of this handing lever, whether fast or slow, is repeated exactly by the moving crosshead of the press and to an extent corresponding with the movement of the handing lever.

There is also no fear of any violent action occurring due to the forging slipping or any accidental cause, as the intensifier cannot possibly overrun, the steam valve being automatically closed towards the end of the stroke. This gear enables very high speeds of working to be safely attained, it being possible, for example, to operate a large press of, say, 6000 tons at a speed of 40 strokes per minute, and taking a smaller size, say 1000 tons press, at a speed of 90 strokes per minute, these figures of course referring to short finishing strokes. A recent improvement is the patent inverted intensifier shown on illustration (fig. 18). Its important advantages are provision against transmission of vibration from the press to the intensifier, increased speed of working due to better control, accessibility of the working parts of the intensifier, and increased stability due to reduction in height. Further important improvements have also been introduced recently in view of the high cost of coal, consisting of patent steam-saving gears for the lifting cylinders and intensifier, the combined effect of which is to reduce the steam consumption of a press by as much as 25 to 40 per cent according to the conditions of working, an important saving in these days of high fuel charges.

Fig. 19 shows a recent type of hot-bloom shear, steam-hydraulically operated, the low-pressure water being provided through the medium of a pre-filler, as in the former instances, at a pressure of 60 lb. per square inch, whilst the steam driver is operated to give the necessary power and movement to the shear. The special feature of this design is that the lower knife rises to cut the bloom. This dispenses with the depressing of the rolls of the shear as in the previous hydraulic types. All moving parts are well guided in massive A frames, and the top knife is kept up by the steam drawback cylinders on either side of the machine. To perform a cutting operation the steam is exhausted from these cylinders, and the top knife is lowered on to the bloom. The main cylinder meanwhile is filled with water from the pre-filler. Now the steam driver is operated, and the moving parts of the shear resting on the bloom cause the lower or cutting knife to rise and cut the same. By releasing the steam driver and admitting steam to the drawback cylinders, the parts of the shear are returned to position for repeat cut.

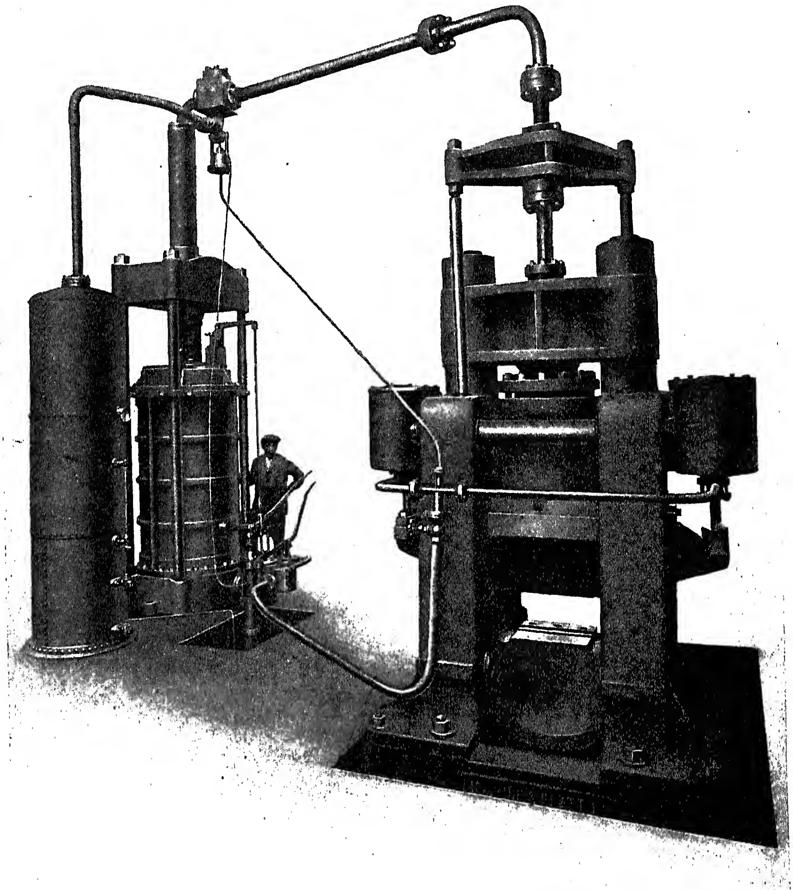


Fig. 19.—Steam-hydraulic Hot-bloom Shear

CHAPTER VII

Oil Presses and Veneering Presses

OIL PRESSES

There are two kinds of press in general use for expressing oil, namely, the Anglo-American Press and the Cage Press. On seeds containing only a minimum percentage of oil, such as cotton seed, linseed, soya beans, and kopak, &c., the former is used; also for second pressing of ground nuts, copra, and other material which it is usual to press twice; while the latter is employed on material containing more than 40 per cent of oil. In both cases the material requires to be previously prepared into meal.

In oil plants it is customary to use the same kind of oil under pressure as is produced, so that the expressed oil shall not be contaminated in case of leakage; and, in addition, in expressing edible oils the greatest care is taken to prevent any leakage escaping into the oil produced.

Fig. 20 shows an Anglo-American press consisting of a base and head joined by four columns. The cylinder and ram are located in the base, the top of which is flanged so as to form a tray to catch the expressed oil and lead it into the strainer on its way to the settling tank. A moving table is secured on the ram, and between the moving table and head corrugated steel plates are fitted, supported by links or racks, so that when the table is in its lowest position the plates are equidistant from each other. The meal, after being heated in a kettle and moulded into cakes, is placed on the corrugated plates and pressed between canvas. The cakes are then removed and their edges trimmed; the trimmings, which contain a high percentage of oil, being returned to the kettle for a second treatment. Sixteen cakes are usually made at each pressing, which is determined by the space required between the corrugated plates for the cakes and the height an average attendant can reach.

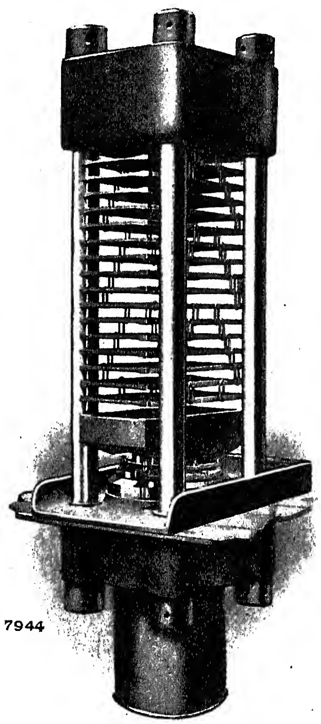

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Fig. 20.—Anglo-American Hydraulic Oil Press

Anglo-American presses are usually arranged in a battery of four, or multiple of four, according to the size of the mill, connected to the same pressure supply but working separately. That is, while one is being charged, the second may have just had the pressure turned on, while in the third the pressure would be standing on, and in the fourth the pressure may just have been released, the descending ram liberating the cakes one by one. In the extraction of stearine the press is similar to the one described, except that the plates are usually plain, while in the extraction of naphthalene the plates are steam heated similar to the veneering press.

Cage Press.—Fig. 21 shows a press of this type, the cylinder with its ram being located below and connected by four columns to the head, which carries a moving top ram (dummy head) mounted upon a slide actuated by rack and pinion. The cage is mounted between the main and top rams concentric with same, and is free to float during the pressing operation, so that the material is pressed equally from each end of the cage. It is also provided with stripping catches. The cage is made square or

round, according to the shape of the cake required, and the rams are ma a loose fit. The cage also requires to be very strong to resist the interr pressure. In the circular for it is built up of vertical st bars, the internal edges which touch each other, a are machined to leave narr slits for the oil to escap the bars being held in po tion by steel hoops. In t square form the inner surf is formed of machined pla perforated with fine holes.

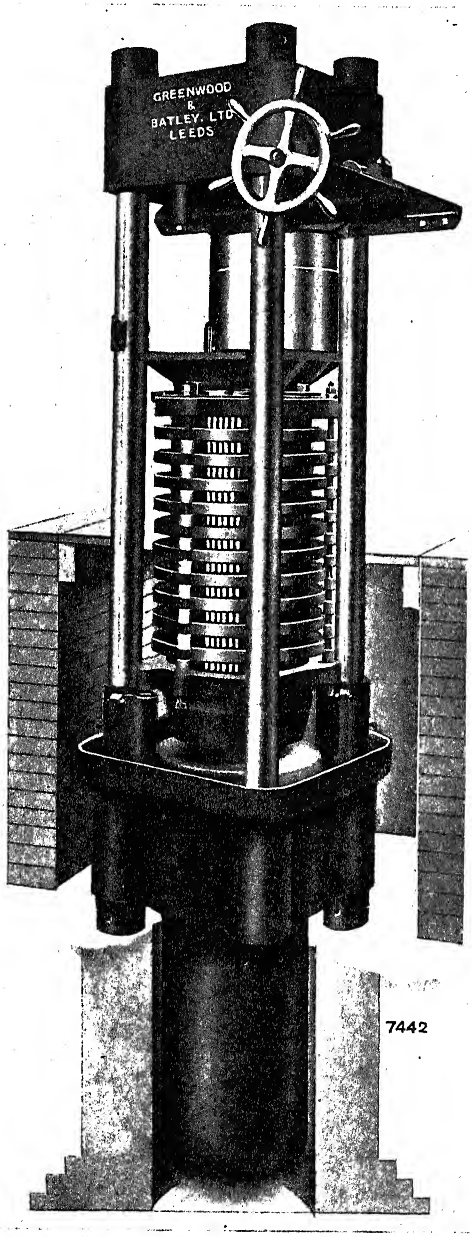

Fig. 21.—Cage-type Hydraulic Oil Press

The method of worki is as follows: as the main r slowly descends, the cage charged with heated or c meal as required from t strickling box, which ho the correct quantity to fo one cake. After each cha of meal is placed in the c a mat is put in, followed b close-fitting plate and anot mat, and so on until the c is filled. The top ram is th brought into position and pressure applied. In lar presses the cage is sometin locked to the ram, in wh case the pressure is relea when the charge has b partly compressed by the ram, which enables an ad tional charge to be insert by this means increasing capacity of the press. T the top ram is again brou over the cage, and, the r catches having been wi drawn, the final pressing commenced, the cage be free to rise, equalizing pressure and thus ensur a uniform yield on all the cakes. After pressing, the top ram is drawn b and pressure applied to the main ram, which forces out the finished c

with the mats and plates, from which they are separated, automatic stops preventing the cage from rising during the operation. The cage is then

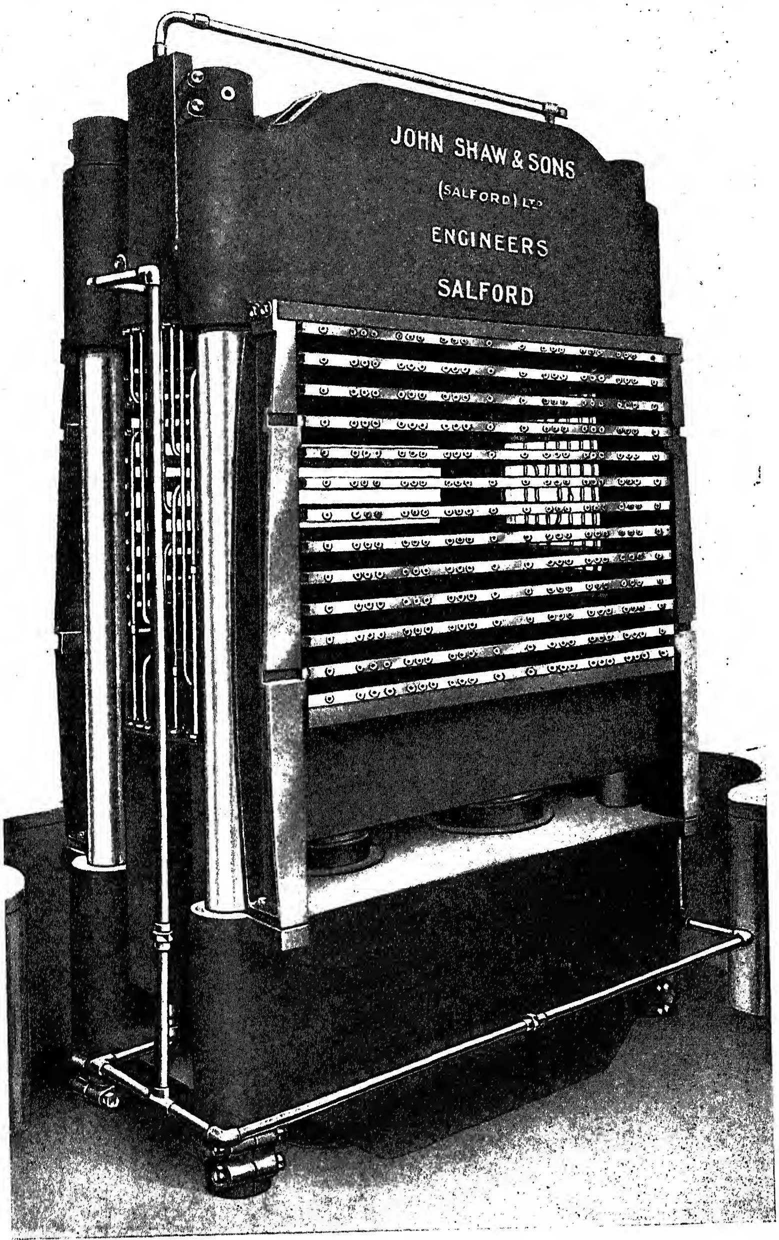

Fig. 22.—Hydraulic Veneering Press

ready for refilling with the main ram in the correct position. It is usual to arrange those presses in pairs, so that one may be under pres-

sure while the other is being emptied and refilled. When a considerable output is required, as many as six finishing presses may be grouped in one battery, served by one or two extractor compressors, whose function is to fill and empty the cages, exerting at the same time a preliminary pressure. As soon as a cage is filled to its utmost capacity it is transported from the compressor extractor to one of the finishing presses by means of a carriage running on rails, and another cage which has been sufficiently long under pressure is returned to the compressor extractor to be emptied of its cakes and refilled with meal. By this system there is a saving in time, labour, and power.

VENEERING PRESSES

The extended use of plywood has caused a considerable demand for large veneering presses, such as shown in fig. 22. The press itself is of the vertical four-column type with one or more cylinders located in the base connected to a common moving table. Between the moving table and head a number of steam-heated plattens are located, provided with links or stops, which separate them an equal distance from each other when open, at the same time allowing them to close freely. The plattens are usually machined steel plates, so drilled that the steam admitted will circulate uniformly, the steam and drain headers being located at the sides, the plates being connected thereto by either flexible tubing or telescopic connections in the manner shown.

CHAPTER VIII

Baling Presses

A simple form of baling press as used for goods such as cloth, hides, &c., consists of a base connected to a head by four columns and a cylinder (usually located in the base) containing a ram connected to a moving table, which is usually made to slide on the columns. The goods are pressed between the head and moving table, which are usually fitted with stillages or plattens grooved to receive the lashing plates, which hold the canvas-covered bale together when removed from the press. When the material to be baled is loose, such as hay, waste, flax, and cotton, &c., a box is used which is either made of timber or metal, according to the density of the bale required, as it must withstand the lateral pressure of the material during the process, which is often considerable at the completion of the operation. Furthermore, the box must be made so that the four sides closely fit the moving table or follower which it guides, and one side must be provided with hinges and catches, in order that the box may be removed on runners and

rails and to enable the bale to be lashed. When an increased output is desired, two boxes are used, one of which can be filled while the other is being pressed.

Presses for baling cotton and jute in large quantities are sometimes very complex on account of the necessity of obtaining as great an output as possible. The ordinary baling equipment consists of two presses, one called the "half press" and the other the "finisher".

Fig. 23 shows the half press with its deep box for holding the loose cotton or other material; made in sections, strongly ribbed, machined on the insides, and furnished with strong hinged doors and locking gear for extracting the bale, also with steel-plate shutters at the bottom for adjusting the gunny bag. The base and head are massive castings connected by four columns furnished with buttress-threaded nuts. Three cylinders with gun-metal shoed rams are located in the base, and are connected to the follower which is machined to fit the box. The follower and head are fitted with stillages. The press is mounted upon two chairs prepared for securing to the foundations, and the control valves are usually fixed on one side.

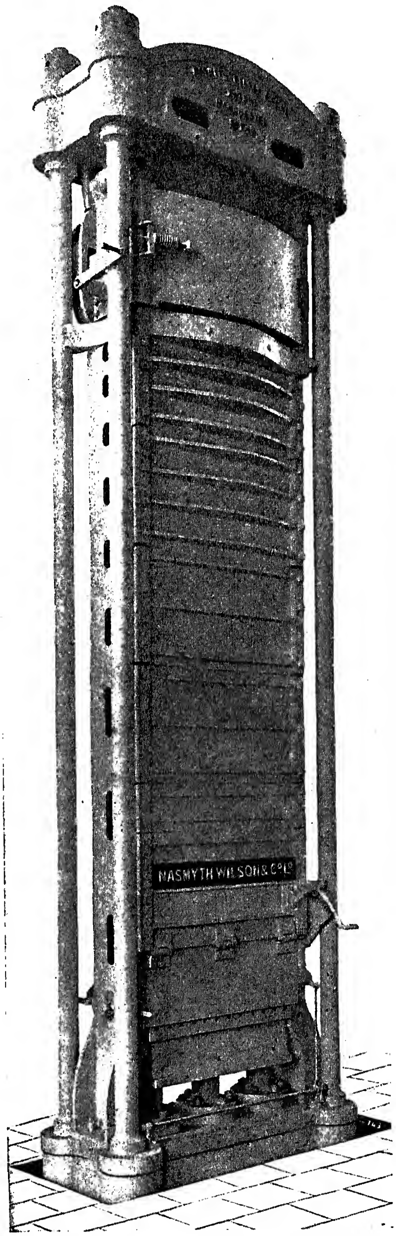

Fig. 23.—Hydraulic Cotton Half Press

Fig. 24 shows the finisher, which consists of a base and head of massive box section, joined by two columns having buttress-threaded nuts, the whole being mounted upon two chairs prepared for securing to foundations. On the base are mounted two cylinders with their rams entirely encased in gun-metal, connected to a common follower guided by the columns, the follower and head being fitted with stillages. The control valves are usually located at the side as in the half press. The extractor, which is not shown, resembles a large pair of tongs. It is carried by a monkey carriage from a rolled-steel joist fixed to or over the half press and finisher, and is provided with

a balance weight and pair of chain blocks. The method of working is as follows: after the box has been filled with cotton or other material, the pumps are started and the full output of low-pressure water is delivered into the central cylinder, while the two side cylinders draw in

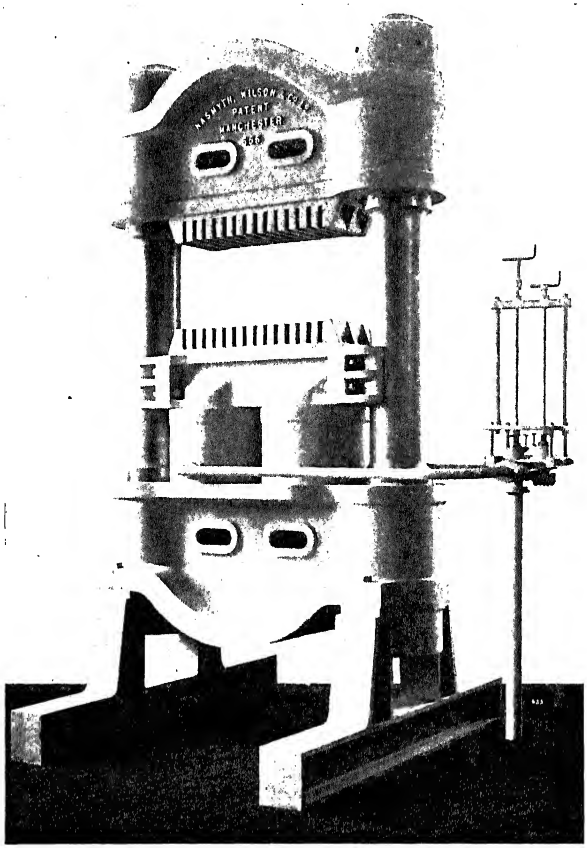

Fig. 24.—Hydraulic Cotton-finishing Press

water through check valves from an overhead supply until the pressure produced almost balances the resistance in the box, which usually takes place at between $\frac{3}{4}$ and $\frac{7}{8}$ of the full stroke; the higher pressure is then admitted to all three cylinders, and the completed bale is then transferred by the extractor to the finisher, where it is subject to considerable pressure and securely lashed.

CHAPTER IX

Cranes and Lifts

Wherever hydraulic power is available and the load is fairly constant, hydraulic cranes have the advantage in the first cost, economy in working, maintenance, reliability, and require less skilful operators than electric cranes. The general impression prevails that hydraulic cranes and lifts are slower than electric. This is not necessarily the case, and the impression probably arises by comparing a hydraulic crane which has been built many years with a recent electric crane, and overlooking the speeding up which has taken place in the meantime.

In cranes and suspended lifts the multiplying device for raising the load the required distance is called a jigger, the multiplying effect being obtained conversely to the ordinary block; one set of sheaves is carried in the crosshead of the ram, and the other at the opposite end of the cylinder. The rope or chain is anchored to the cylinder at one end, passes round the sheaves, and is attached to the cage or crane hook at the other end. Adjustable stops are provided on the crosshead guides, rendering over winding impossible, and at the same time a simple tappet gear automatically brings the control valve into the closed position and the load gradually to rest.

Jiggers are used on all three crane motions, namely, hoisting, luffing, and slewing, each operated by a hand-lever control valve having three positions. The control is therefore simple and can be left in the hands of an unskilled man. Furthermore, the motions are silent, have a great elasticity in speed from creeping to the maximum speed according to the position of the hand lever without the use of a brake.

Flexible steel-wire ropes may be said to have taken the place of chains except for slewing, being cheaper, lighter, and running more sweetly over barrel and pulleys. The earliest form of lift is that known as the "direct" type, in its simplest form consists of a cylinder, ram, platform, guides, and control valve. All the working parts are below the platform, the ram being secured to it in such a way as to give rigidity and strength necessary for dealing with roughly handled loads. In direct lifts for heavy loads additional rams are sometimes employed. To save power the centre ram is coupled to the power main without valves and proportioned so as to nearly balance the empty lift. Two additional rams are located, one at each side of the centre ram. These take the load, and are provided with a common control valve. If the load is a variable one, there may be four or more lifting rams coupled in pairs, so controlled that either or both pairs may be employed according to the load, a check valve being fitted to the supply to the control valve or valves to prevent accident in case of a burst main.

Fig. 25 shows a pair of wagon tippers of the direct oscillating ram type, such as are intended to deal with railway wagons having a hinged door provided at one end, without having to turn the wagon round. The wagon

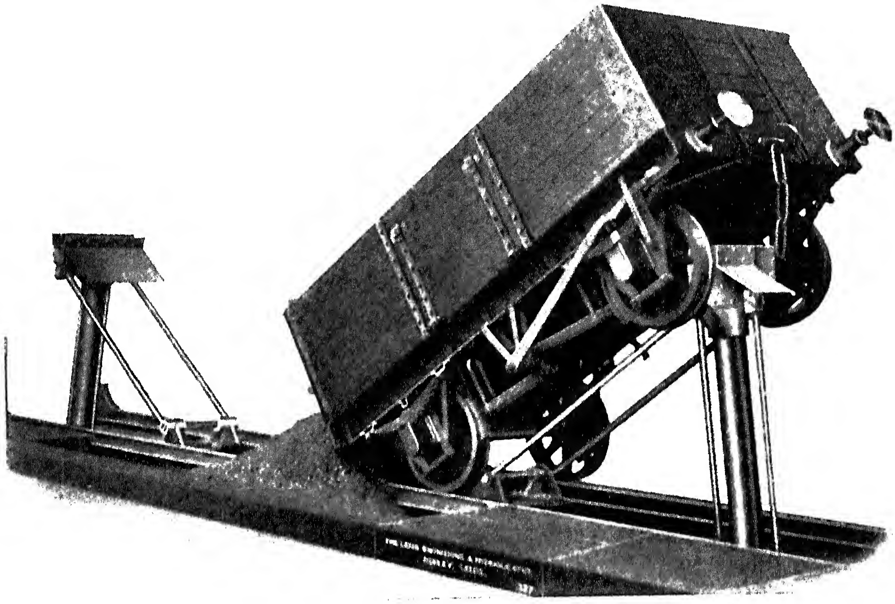

Fig. 25.—Hydraulic Railway-wagon Tipper

having been placed in position, the tipper remote from the hinged door takes hold of the axle and tips the contents of the wagon into the hopper as shown, then lowers the wagon back on to the rails and recedes until the head is below the rails. Sometimes the tipper is of the fixed-cylinder type,

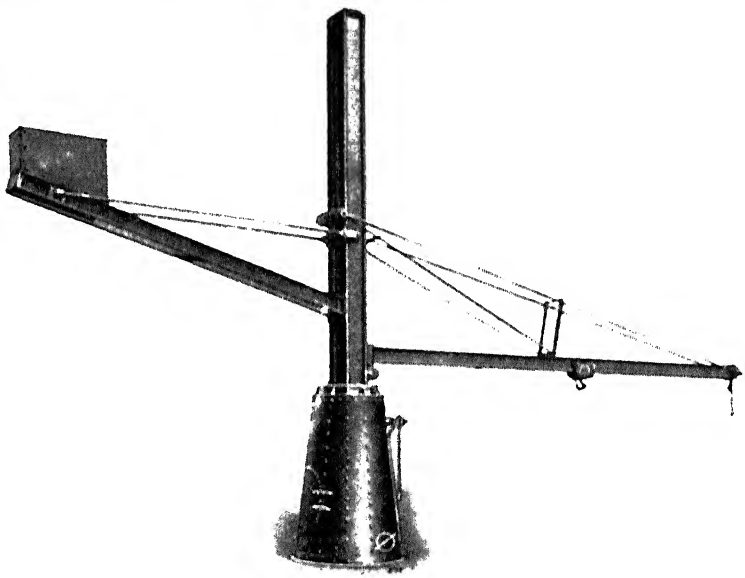

Fig. 26.—Hydraulic Direct-lift Fixed-pedestal Crane

located between two hoppers which it serves. In the case of long wagons, in order to obtain the required angle which is determined by the buffers fouling the rails, the ram is attached to a hinged platform on which the wagon is run and secured.

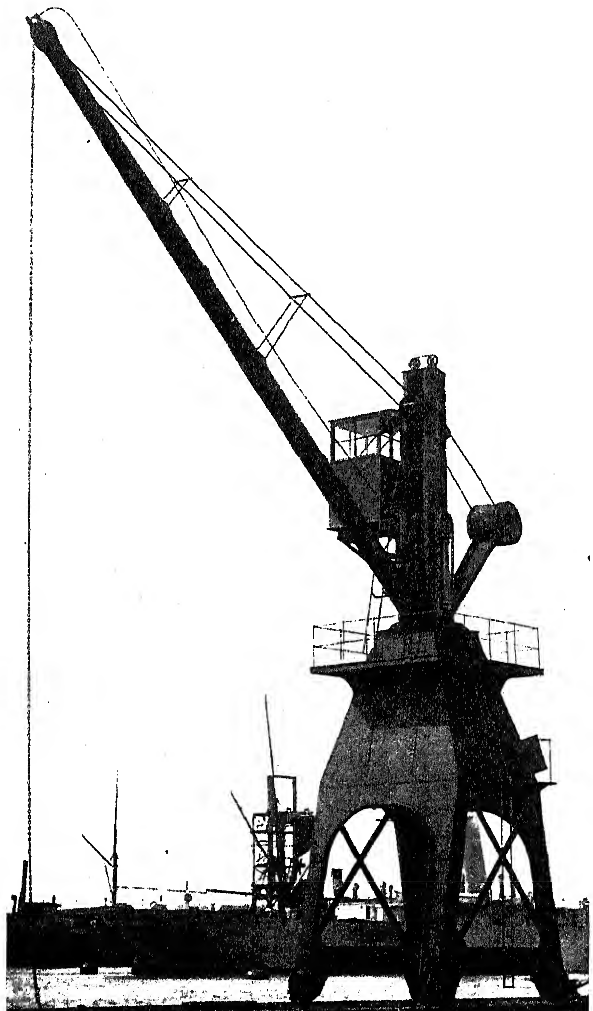

Fig. 27.—Hydraulic Movable Portal Crane

Fig. 26 shows a direct-lift fixed-pedestal crane. It is frequently found in shipyards and structural steel works, and seldom exceeds a power of 5 tons and 10 ft. lift, but with varying rake up to 40 ft. It will be seen that the horizontal jib with its tension rods is attached to a carriage which runs on the mast and is directly coupled to a lifting ram. The mast supports and monkey runners are usually mounted upon ball or roller bearings so that

the load may easily be manually moved in any direction. The mast is provided with a counter-balance arm, and the absence of chains and ropes makes its construction exceedingly simple.

Fig. 27 shows a movable portal crane with fixed rake but of the two-powered type, the opening in the pedestal being large enough to allow a

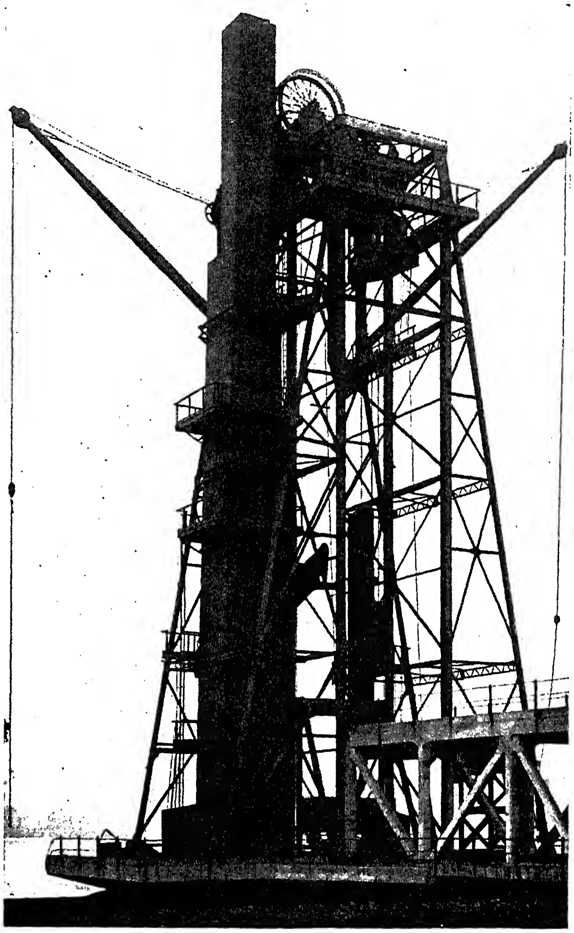

Fig. 28.—Hydraulic Coaling Hoist

wagon to pass through loaded up to gauge limit. This crane moves on rails by manual power. It will be observed that the cabin is fixed on the mast, so that the driver has a complete view of the load under all conditions. In movable cranes the connections are made by flexible pipes, mostly of the armoured hose type.

Fig. 28 shows a coaling hoist of the suspended type, capable of dealing with wagons having a gross load of 30 tons. The framing is constructed of steel plates and angles with central guides, between which the cradle

works. The cradle, which is provided with a tipping platform, is suspended by four steel-wire ropes passing over the conveyance sheaves at the top of the framing to the hydraulic lifting machinery, which is attached to the side of the framing, and cased in as shown in illustration. This machinery consists of three hydraulic cylinders fitted with rams working downwards. The centre cylinder is in constant communication with the accumulator pressure, so as partially to balance the weight of the cradle. The tipping cylinder consists of a hydraulic cylinder placed above the lifting cylinders, and is fitted with two steel-wire lifting ropes. The lifting and tipping ropes both work over the lifting crossheads, but only the tipping ropes work over the tipping crosshead, the arrangement being such that the tipping frame on the cradle can be tipped at any point in the lifting range. The shoot is of steel and 33 ft. 6 in. long, giving an out reach of 27 ft. beyond the framing when lying at the usual angle for shipping coal. This shoot is fitted with two screens and doors for controlling the flow of coal. A 5-ton antibreakage and a 3-ton screening crane are provided on each side of the framing, as shown in illustration. The loaded wagons are brought on to the hoist at the low level, and when discharged are taken off at the high-level gantry. Hydraulic hauling jiggers are provided for this purpose.

CHAPTER X

Capstans

The usual form of capstan for goods yards, quays, &c., for hauling railway wagons is the turnover type, shown in its working position in fig. 29 and in the inverted position in fig. 30, in which the working parts are accessible It is capable of exerting a pull of a ton, and is operated by a pedal valve

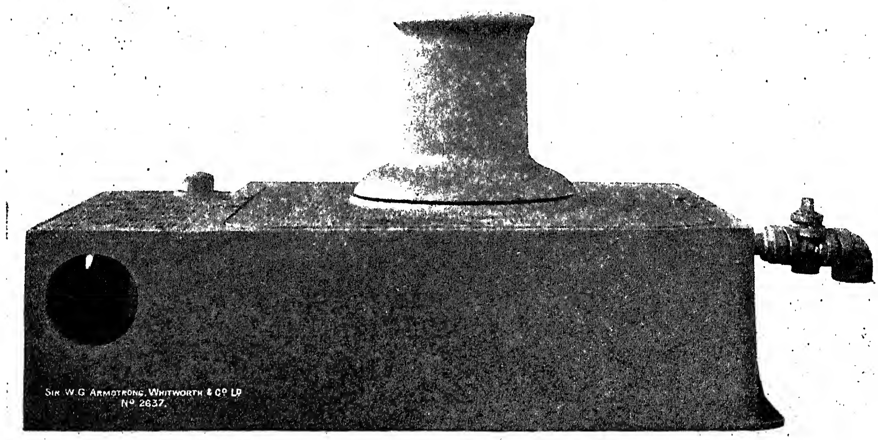

Fig. 29.—One-ton Hydraulic Turnover Capstan

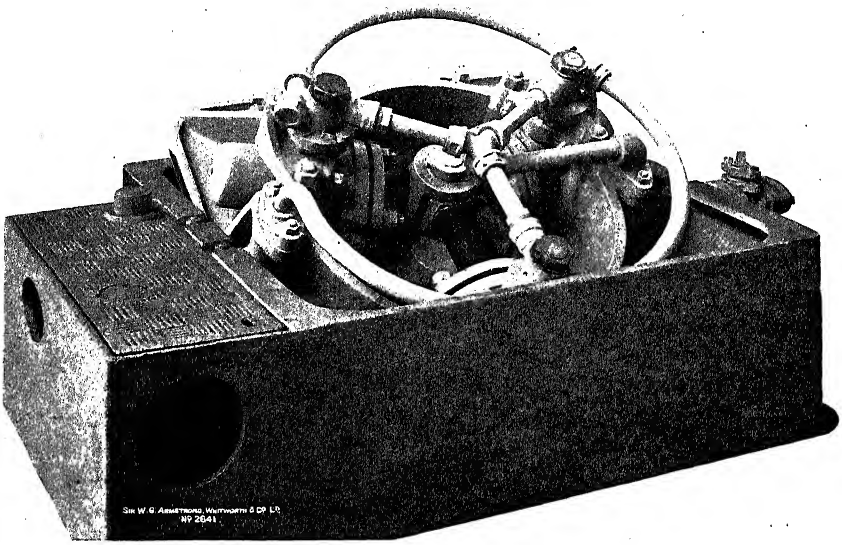

Fig. 30.—Turnover Capstan (inverted position)

As will be seen, there are three single-acting rams direct coupled upon a common crank-pin. The oscillating motion of the cylinders operates the working valves, which are of the well-known "trunnion" type. Fig. 31 shows a double-power geared capstan of 5/3 tons power, in which the change of speed is obtained by making the head of two diameters. The control valve in this case is of the screw-down type. The engine is of the quadruple-acting ram type, and the working valves are of the balanced-beat type highly suitable for high pressures.

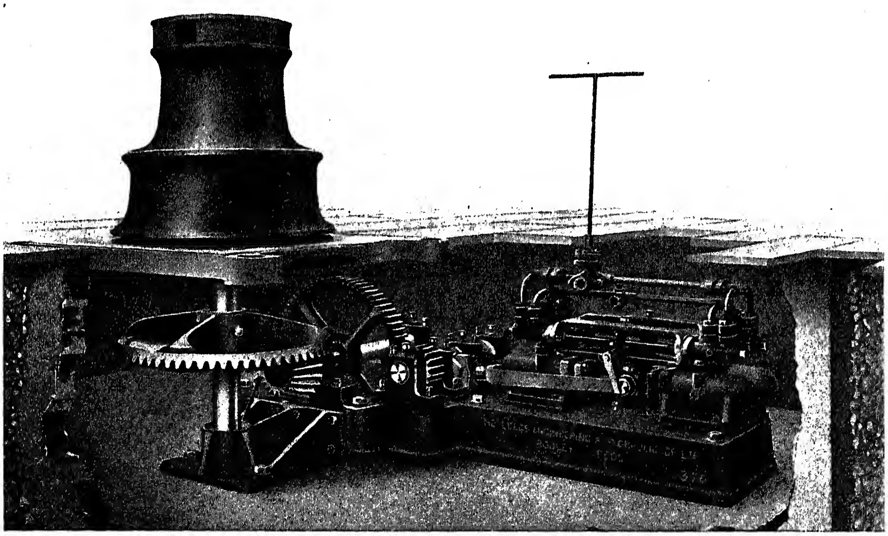

Fig. 31.—Hydraulic Geared Two-power Capstan

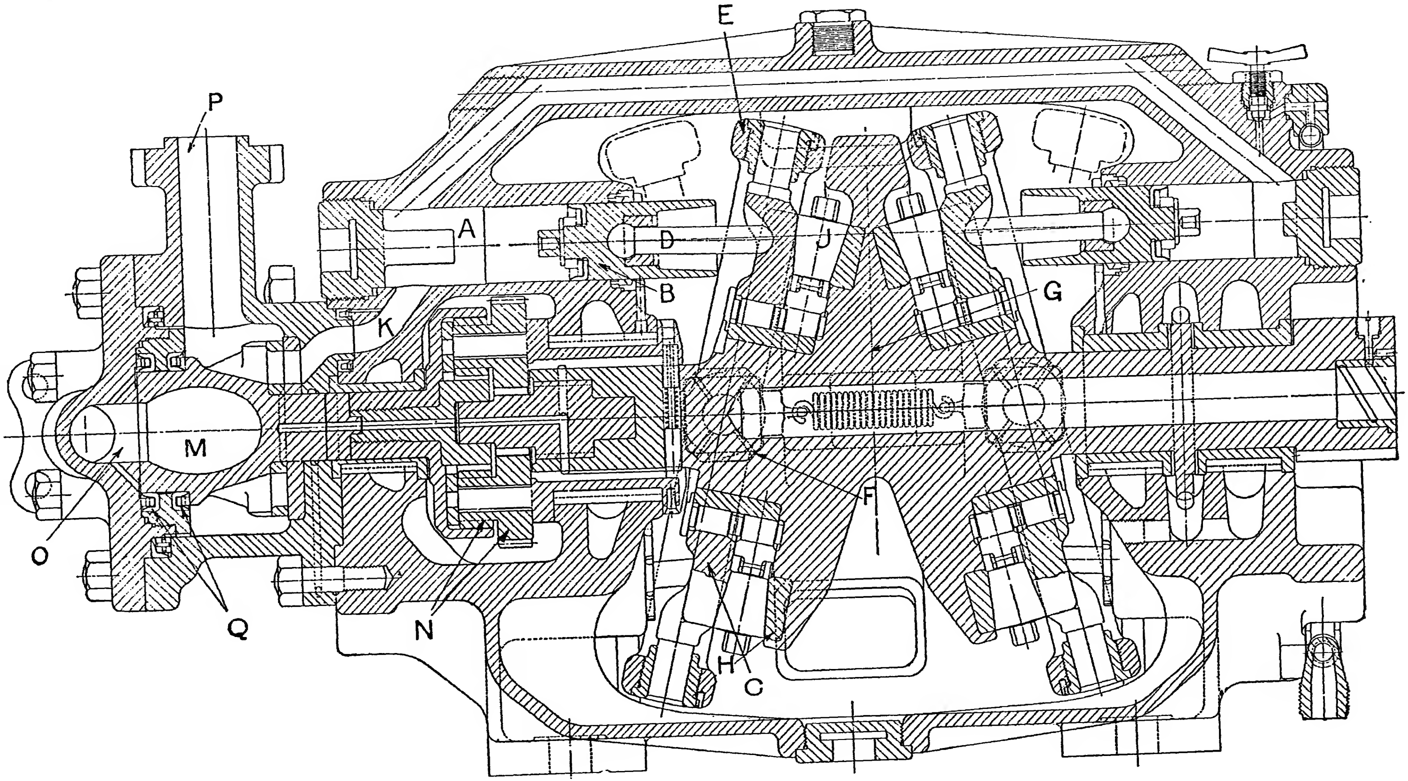

Fig. 32.—Sectional Drawing of a Hydraulic Swash Plate Engine

CHAPTER XI

Swash Plate Engine

Hydraulic power has been used in battleships for a number of years for a variety of purposes, the most exacting being the training of guns, in which great range of speed, uniform turning moment, and ability to start loaded in any position has led to the development of the swash plate engine which appears to fulfil these exacting requirements. In this type of engine the reciprocating motion of the piston is converted to the rotary motion of the shaft by means of the swash plate, which is a revolving circular plate set obliquely to the axis of rotation, the plate and shaft being in one piece. The principle of the machine is that the pressure exerted on an inclined surface in a direction parallel to the axis produces a rotary action of the swash plate and shaft.

These engines are constructed with either single or double swash plates, the former having one set of cylinders operating on one swash plate, while the latter has two swash plates set back to back and two sets of cylinders, so that balancing pressures are exerted by the pistons and the swash plates. Seven cylinders, arranged in a circle round the shaft, are provided for each swash plate. In the following description the reference letters refer to fig. 32, which is a sectional arrangement of a double swash plate engine. The pistons B act upon the rocking swash plate C, by means of connecting rods D. The swash plate C is mounted by means of trunnion bearings, in a gimbal ring E; this gimbal ring in its turn is mounted on trunnion bearings F formed in the casing. The revolving swash plate G consists of a circular inclined plate formed in one with the main shaft of the engine. One face of the inclined plate is mounted on a hardened roller path H. Roller or ball bearings J are fitted between the rocking swash plate C and the revolving swash plate G. Pressure is admitted to the cylinders A by ports K, the flow of water being regulated by valve M, which is rotated by gear wheels N, driven by the engine shaft and in an opposite direction to that of the shaft. Valve M, which is rotated at one-sixth the speed of the engine, is provided with twelve ports equally spaced, six for pressure and six for exhaust alternatively; and during one revolution of the engine each cylinder coincides with a valve pressure port and an exhaust port consecutively. Of these twelve ports six communicate with the pipe O through the interior of the valve, and six with pipe P through the exterior of the valve, the interior and exterior passages being separated by means of cup leathers Q, so that by using pipe O for pressure and pipe P for exhaust, or conversely, the direction of rotation of the engine may be changed. With this engine a very even torque is obtained, and it can be run at an exceedingly slow rate without danger of pulling up, which was a source of trouble with the types previously used.

The author desires to express his indebtedness to the following companies for the use of illustrations:—The Leeds Engineering & Hydraulic Co., Ltd.; Messrs. Rice & Co. (Leeds), Ltd.; Messrs. Fielding & Platt, Ltd.; Messrs. Greenwood & Batley, Ltd.; Messrs. Sir W. G. Armstrong, Whitworth, & Co., Ltd.

WATER TURBINES

BY

A. H. GIBSON

D.Sc., M.Inst.C.E., M.I.Mech.E.

Water Turbines

Impulse and Reaction Turbines.—Modern turbines may be divided into two classes, Impulse and Reaction turbines. Of the former the Pelton wheel, and of the latter the Francis turbine, or one of its modifications, are the only types used in recent important installations.

In an *Impulse turbine*, the whole head of the supply water is converted into kinetic energy before the wheel is reached. The water leaves the nozzle or nozzles in one or more high-velocity jets which are exposed to the pressure (usually atmospheric) obtaining in the turbine casing. It then impinges on a series of buckets carried by the wheel, and in virtue of the change of direction and hence of tangential momentum produced by these buckets, exerts a driving force and so does work on the shaft. Its direction is freely deviated by the buckets, and its pressure remains uniform during its passage through the turbine.

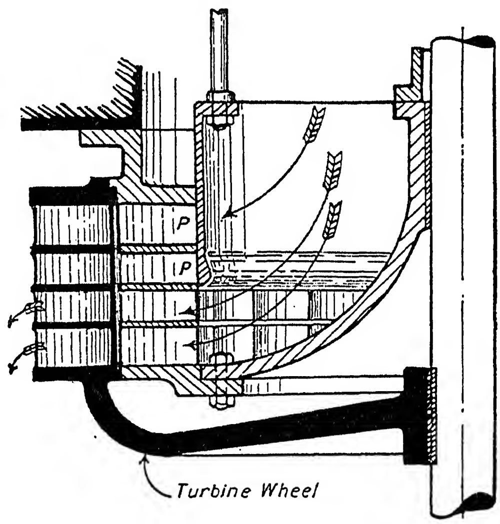

Fig. 1.—Section through Fourneyron Turbine, with Inside Cylinder Gate Regulation

The *Pressure* or *Reaction turbine* consists essentially of a wheel or runner provided with vanes into which water is directed over the whole periphery by a series of guide vanes. The water on leaving these guide vanes is under pressure, and supplies energy partly in the kinetic and partly in the pressure form. In its passage through the runner the pressure energy is utilized in increasing the relative velocity of flow between the vanes, and the water finally leaves the runner at the pressure obtaining in the discharge pipe or draft tube.

In the earliest of these turbines, the Fourneyron, the guide vanes P were inside the runner, forming an outward-flow turbine (fig. 1). This was followed by the Jonval turbine, in which the guide vanes are above the runner

and the water flows axially into and through the wheel, giving an axial-flow turbine (fig. 2). Both these types have been to all intents obsolete for some years, and the Francis or inward-flow turbine, in which the guide vanes surround the outer periphery of the runner, is now in general use. In the earlier Francis turbines the discharge was also radially inwards, but in modern turbines, in order to obtain a larger discharge area with a given diameter of runner, the form of the buckets has been modified so as to give a discharge in a direction which is more or less parallel to the axis of the turbine. In the most recent turbines for low-head plants, the design is indeed tending to a turbine which, as regards the runner, is essentially of the Jonval or axial-flow type (fig. 3). Inward-flow guide vanes are used, but there are signs that even in this respect the design of low-head turbines is tending to revert to a modified Jonval type.

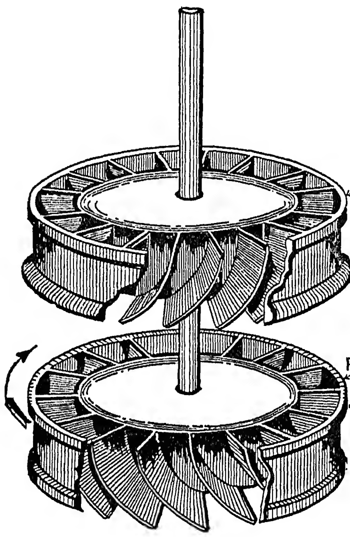

Fig. 2.—Guide Vanes and Runner of Jonval Turbine

General Comparison of Impulse and Reaction Turbines.—The peripheral velocity of a Pelton wheel for maximum efficiency is slightly less than one-half the spouting velocity of the jet (usually approximately $0{\cdot}46\sqrt{2gH}$, where H is the head), while that of the reaction turbine varies from about $0{\cdot}65\sqrt{2gH}$ to $1{\cdot}05\sqrt{2gH}$, depending on the design. Because of this, the Pelton wheel is well adapted for very high heads, which may then be utilized with moderate speeds of rotation. On the other hand, the relatively high speed of the reaction turbine enables reasonably high rotative speeds to be obtained with low heads.

The Pelton wheel cannot well be designed to utilize efficiently more than two jets on a single wheel, and as the maximum practicable jet diameter is about 12 in., the volume of water which can be handled and the output of the turbine become small under low heads. The reaction turbine with its full peripheral admission is well adapted for large volumes. It is not suited for small powers under high heads, since the volume of water is small, the waterways are of very small sectional area and easily become choked by floating debris, and the fluid friction losses become relatively high.

The Pelton wheel is not well adapted to be used with a suction or draft tube, and, where the tail-race level varies appreciably, must be installed above the highest tail-water level with some sacrifice of head. The reaction turbine lends itself readily to this construction, and has the further advantage that it may be drowned without loss of efficiency. The efficiency of the reaction turbine is not so sensitive to changes of head as that of the Pelton

wheel, and since the percentage variation in head is usually greater in low-head than in high-head plants, this is another reason why the Pelton wheel is not well adapted for low heads.

If operated under constant head and constant speed, the efficiency of

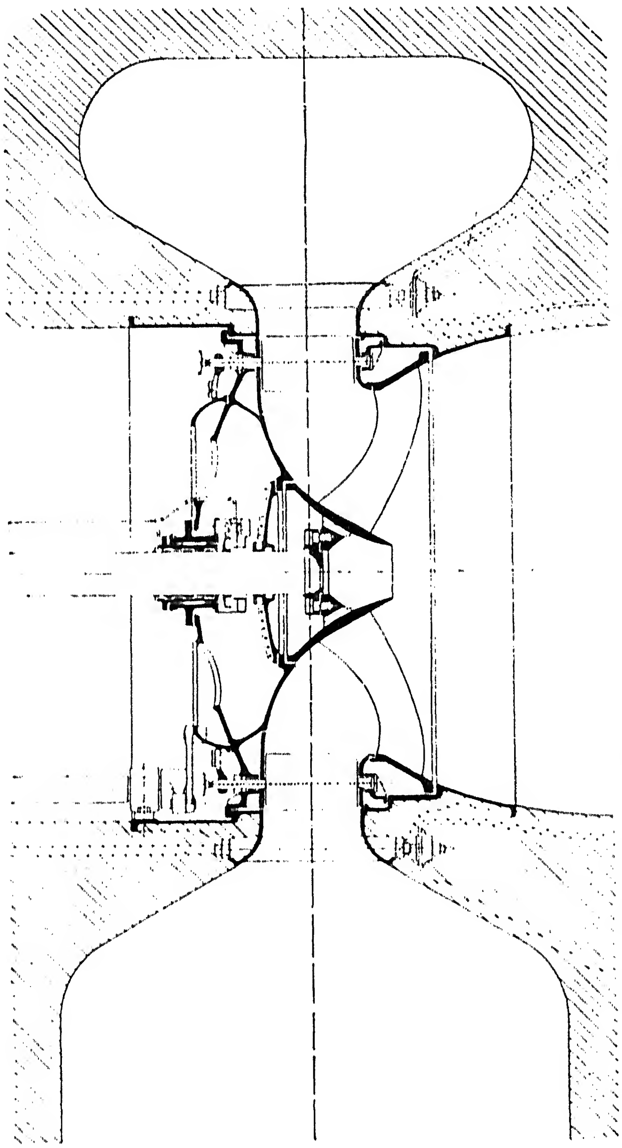

Fig. 3.—Modern High-speed Francis Turbine giving 10,000 h.p. under 10 metres (32·8 ft.) head at 93·7 r.p.m. Specific speed, 118·5. Efficiency at full load, 85·0 per cent

the Pelton wheel does not fall off so rapidly at part loads as that of the reaction turbine. On the other hand, the modern reaction turbine has a slightly higher full-load efficiency, so that the average efficiency from half to full load is sensibly the same in a well-designed machine of either type. The following table shows typical values of the part-load efficiencies

of modern turbines of both types of large size, installed under equally favourable conditions.

Proportion of maximum discharge ..	0·2	0·3	0·4	0·5	0·6	0·7	0·8	0·9	1·0
Efficiency of reaction turbine	0·60	0·70	0·75	0·79	0·82	0·87	0·90	0·91	0·89
Efficiency of Pelton wheel	0·70	0·78	0·82	0·83	0·84	0·85	0·86	0·85	0·83

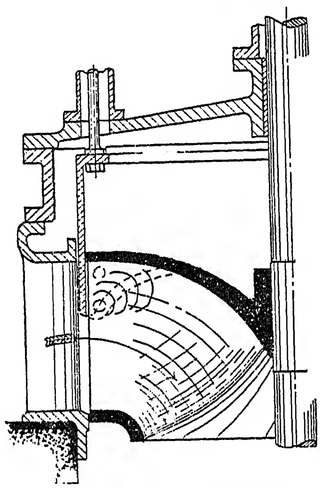

Fig. 4.—Cylinder Gate for Francis Turbine

While efficiencies as high as 93 per cent are on record for large reaction turbines, such values can only be attained by the most careful attention to the design not only of the turbine but also of its setting, and in general the full load efficiency does not exceed 85 per cent in the case of a large reaction turbine, and 80 per cent in the case of a large Pelton wheel. The possibilities of accurate speed regulation are about equal in the two types.

For large units the reaction turbine is generally preferable for heads up to 400 ft. For heads above 750 ft. the Pelton wheel is more suitable, while between these limits the choice depends largely upon local circumstances and on the power required. The greater simplicity and accessibility of the parts requiring replacement due to natural wear and tear renders the Pelton wheel more suitable when the supply is taken from a stream carrying an appreciable amount of grit or silt in suspension.

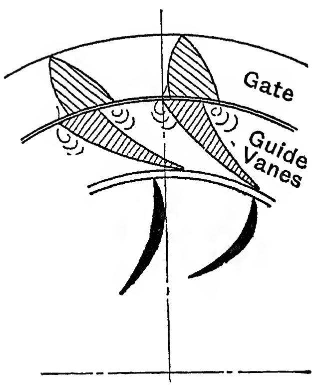

Fig. 5.—Register Gate

The Reaction Turbine: Constructional Details.—The supply of water to the runner of the reaction turbine is regulated either by a cylinder gate, a register gate, or by pivoted guide vanes. The cylinder gate (fig. 4) consists of a plain cylinder sliding axially between the guide vanes, which are fixed, and the runner. Its axial position is regulated by the governor. The register gate (fig. 5) consists of a cylinder carrying appropriate waterways, and capable of rotation about the axis of the turbine. It is usually fitted outside the guide-vane ring.

With either type excessive eddy formation is produced at part gate,

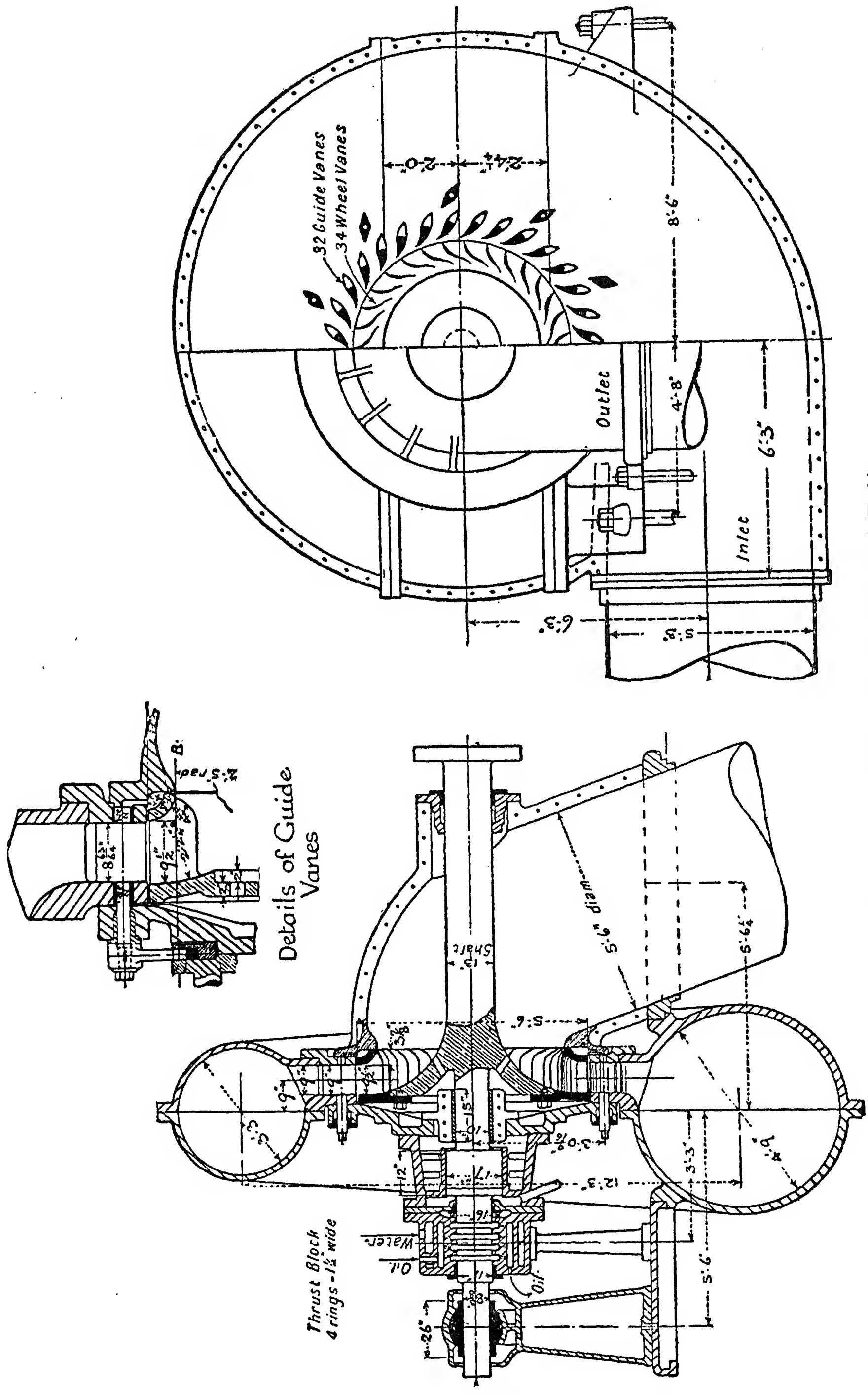

Fig. 6.—Section of Slow-speed, High-head Francis Turbine

giving low part-load efficiencies. While the register gate is practically obsolete, the cylinder gate is still used in small plants having a fairly constant load, and where low part-gate efficiency is unimportant. It is cheap and not so easily deranged as the wicket gate, as the system of pivoted guide vanes (fig. 6) is termed. The latter arrangement is, however, the only one now used in important installations.

In a low-head installation the turbine may be erected in the open forebay, as shown in fig. 16. This method has the disadvantage that the guide-vane mechanism is submerged, and cannot be inspected or repaired without draining the wheel pit, and in most recent important medium- and low-head installations the guide-vane ring is surrounded by a spiral volute chamber, from which the pressure water is delivered with uniform velocity around the entire periphery of the guide ring.

The velocity in this volute ranges from $0{\cdot}15\sqrt{2gH}$ to $0{\cdot}25\sqrt{2gH}$, the higher value applying to low-head plants, and the lower to heads exceeding 300 ft.

For heads not exceeding about 100 ft., modern practice has favoured the moulding of the volute chamber in the concrete of the substructure (fig. 7).*

For higher heads, considerations of strength necessitate a metal casing, which may be of cylindrical section, but which, for single-runner machines, is of spiral volute form (fig. 6). This may be of steel plate, cast iron, or cast steel. Owing to the risk of flaws in the casting and of its unsuitability for withstanding sudden shocks, cast iron is not very suitable for large casings subject to high heads and liable to water-hammer shocks.

Guide Vanes.—The guide vanes or gates are commonly made of cast steel. The stems may be cast in one piece with the vanes, or, in large units, may be keyed to the vanes so as to facilitate the removal of worn vanes. Under very high heads where the water carries an appreciable amount of grit, bronze guide vanes are advantageous for resisting erosion.

The stems project through stuffing boxes in the turbine casing. Each stem carries a lever which is coupled to a common regulating ring concentric with the turbine shaft (figs. 6 and 7), whose position is regulated by the governing mechanism, so that all the guide vanes are opened or closed simultaneously.

The gate stems should be strong enough to resist the stress which would be produced in the case of an obstruction between two vanes with the full effort of the governor concentrated upon them. The links between the levers and the regulating ring should be the weakest part of the system.

Turbine Runners.—For turbines of moderate size under low and moderate heads, the iron runner, cast in a single piece, is general. In order to facilitate shipment and to provide greater assurance of sound castings in very large machines, the runner is usually cast in four quadrants, which are tied together by a heavy cast-steel crown at the top, and by a cast-steel ring which is a force fit around the lower band of the runner (fig. 8).*

For runners of the types shown in fig. 9 B and C, steel-plate vanes

* By courtesy of the Cramp Shipbuilding Company, Philadelphia.

may be used with advantage. These are pressed or hammered to shape against a former, and are cast into the turbine crowns. They have the advantage of being thinner and smoother than the cast vane.

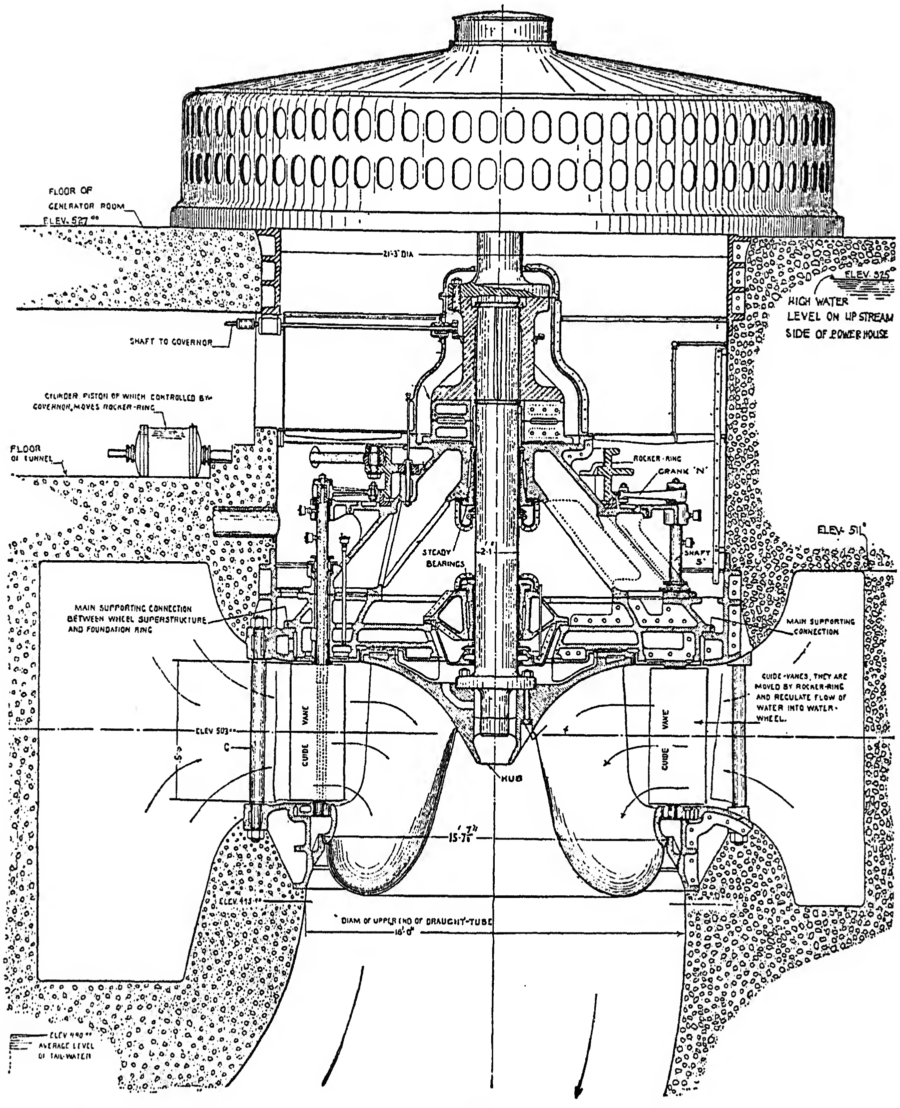

Fig. 7.—Keokuk Turbines of Mississippi River Power Company. Overload capacity, 13,000 h.p.

In the early development of turbines for high heads, considerable difficulty was experienced due to corrosion and wastage of the runner, even with water free from solid material in suspension. It is now generally accepted that such corrosion is due to faulty design of the runner vanes, leading to excessive eddy formation. The pressure at the core of such eddies is com-

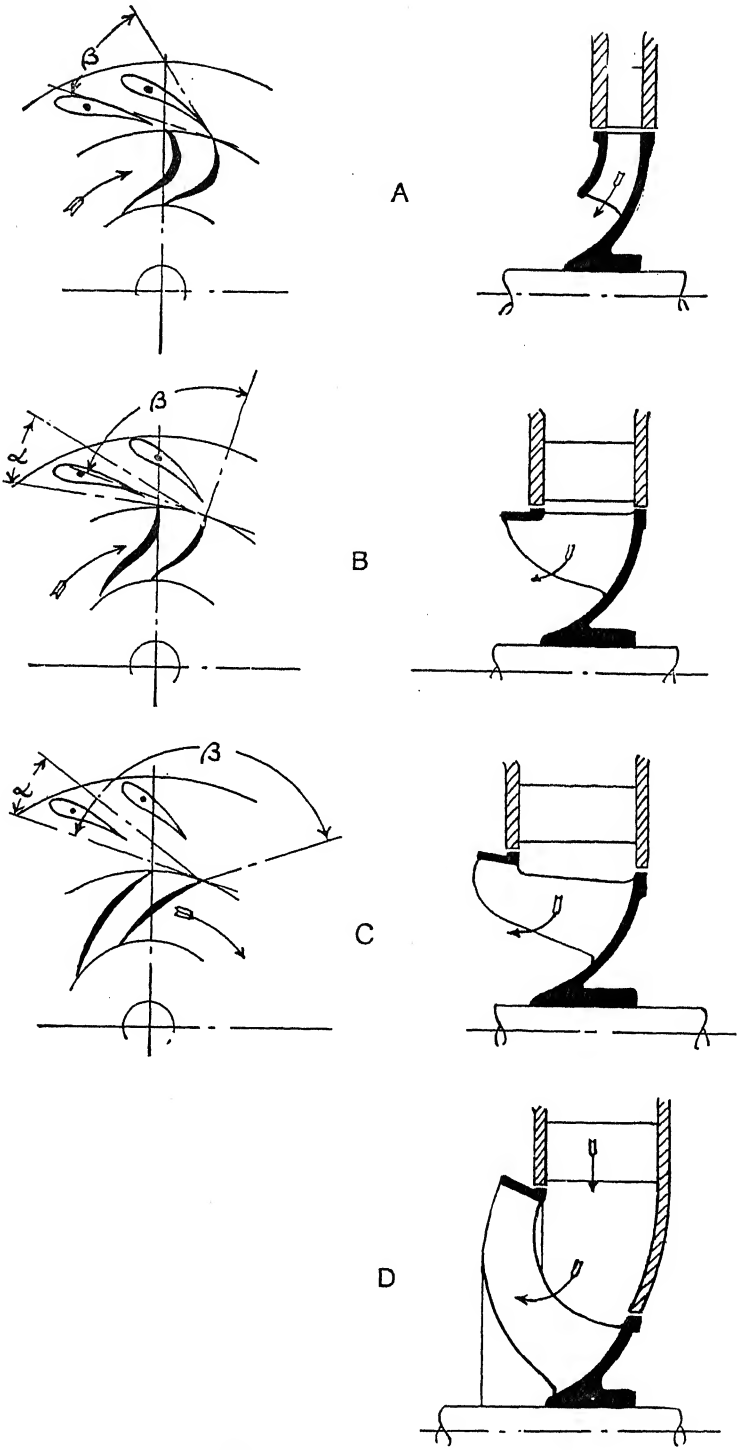

Fig. 9.—Types of Turbine Runner

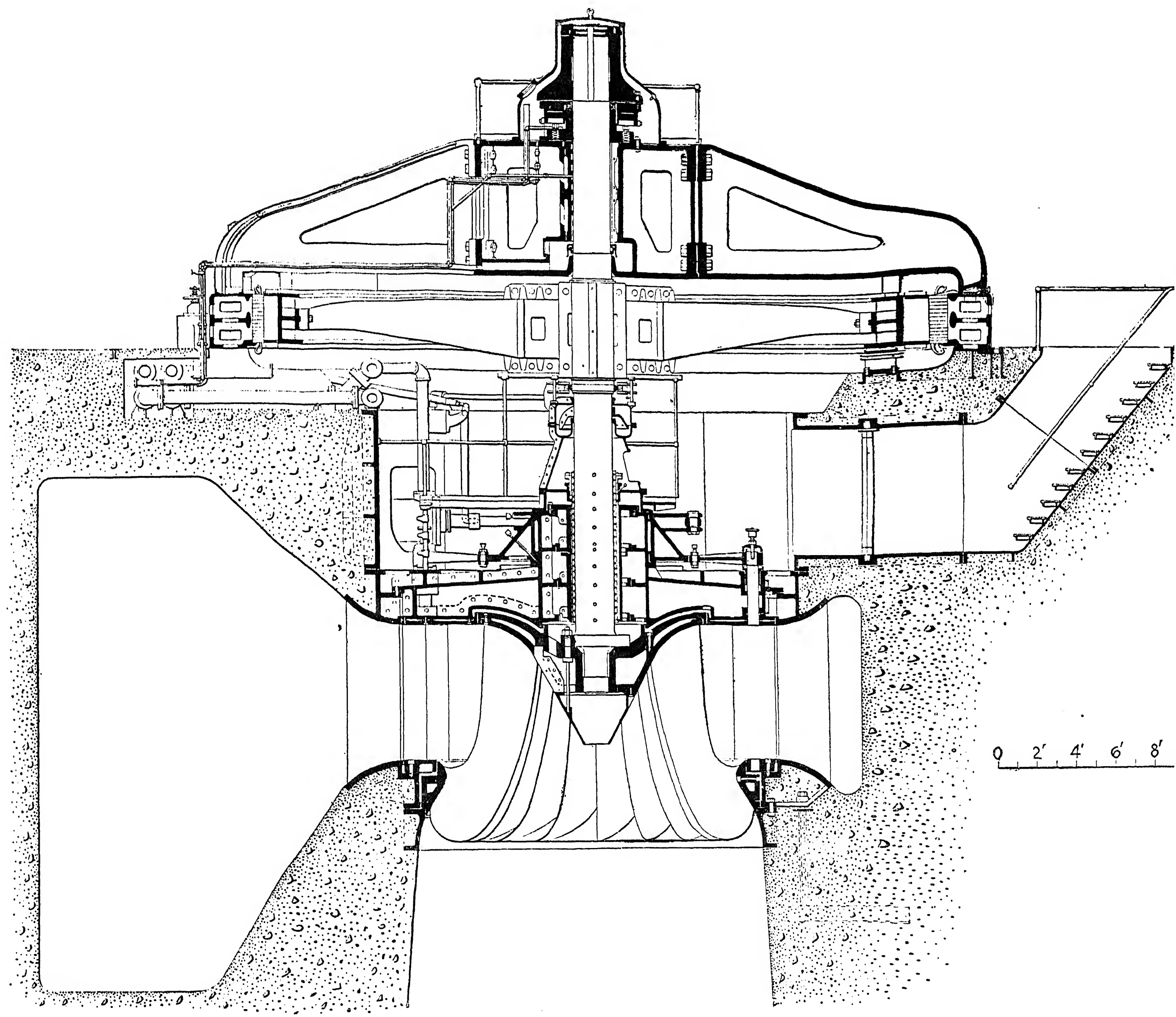

Fig. 8.—CEDARS RAPIDS MANUFACTURING AND POWER CO.

18,000 h.p.; 17 ft. 8 in. diam.; 30-ft. head; 55·6 r.p.m.

paratively low, and this leads to the liberation from solution of air apparently containing excess oxygen, which rapidly attacks and pits any iron or steel surface with which it may be in contact. Such corrosion can largely be eliminated by correct design. Where the supply water is clean, cast-iron runners are to be preferred as being smoother than cast-steel, except where the speed is so high as to necessitate cast steel being used to withstand the centrifugal stresses. Corrosion is pronounced at high heads if the turbines are operated much at part gate, while the practice of running for long periods without load is especially bad for the runners. Where the water carries grit or sand in suspension the runner is usually made of cast steel if large, and of phosphor bronze if small.

The general changes in the shape and proportions of the runner which have accompanied the recent development of the high-speed turbine are indicated in fig. 9, and in fig. 3, which shows diagrammatically one of the latest types of low-head turbine.* The change has been in the direction of increasing the depth of the buckets, and at the same time of maintaining or increasing the ratio of the discharge area at exit to that at entrance. Also whereas it was formerly considered essential that the space between the guide vanes and the runner should be reduced to a minimum, the most recent low-head turbines show a very large radial clearance. The type shown in fig. 9 A is adopted for high head and comparatively small volumes; B is suitable for medium speeds; C for high speeds and medium heads; and D for high speeds and low heads.

End Thrust on Shaft.—With a single-runner turbine, owing mainly to the static pressure behind the runner due to leakage between the runner and the casing, there is, unless special means are adopted to prevent it, an unbalanced end thrust on the shaft.

Where the head is low, the pressure behind the runner may be relieved by a series of vent holes through the runner crown. In order to prevent the water behind the runner whirling and so increasing in pressure outwards due to centrifugal action, a number of radial vanes, almost touching the wheel, are carried by the turbine casing. Any force still unbalanced is taken up by a small thrust block. Where the head is high this method is inadequate, and with a large turbine the end thrust is so large as to make the provision and maintenance of a suitable mechanical thrust bearing a matter of some difficulty. For this reason the greater part of the end thrust is balanced by hydraulic rather than mechanical means. One method of doing this is shown in fig. 6. The space to the right of the balancing piston is supplied with pressure water from the penstock through a regulating valve, while the space to the left is freely vented to the draft tube. Since there is a small leakage past the piston, the pressure in the balancing chamber may be closely regulated to suit the conditions of operation by adjustment of the regulating valve.

In a second method the areas of the runner at A and B (fig. 10 *a*) are made equal. Leakage from the spaces at A and B is reduced to a minimum

* See also Kaplan, *Zeitschrift für des Gesamte Turbinenwesen*, 20th November, 1919.

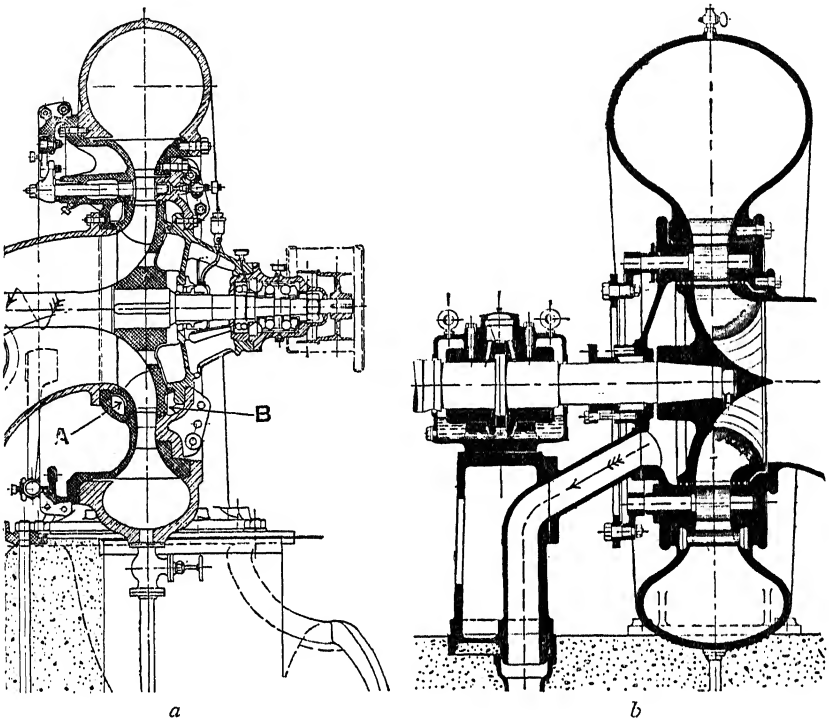

Fig. 10.—Balancing Devices for Turbine Runners

by cutting down the radial clearance between runner and casing, and by the use of labyrinth joints. Any unbalanced force is taken up by a small thrust bearing.

In a third method, the space behind the runner is vented to the draft tube by a pipe

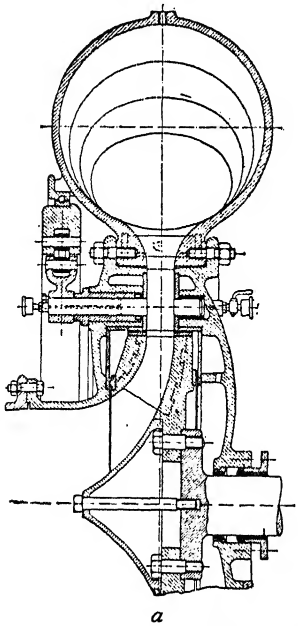
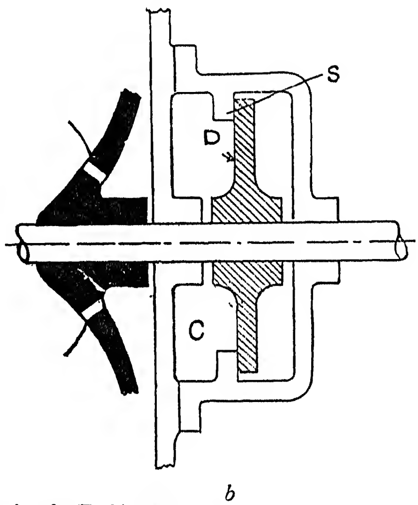

Fig. 11.—Balancing Devices for Turbine Runners

of sufficiently ample dimensions to ensure the pressure in this space being sensibly the same as that in the draft tube (fig. 10 *b*).* Here again, any appreciable wear of the runner may render the discharge pipe inadequate in area, giving rise to an unbalanced force for which a thrust block must be provided.

Fig. 11 *b* shows a balancing arrangement in which no thrust block is

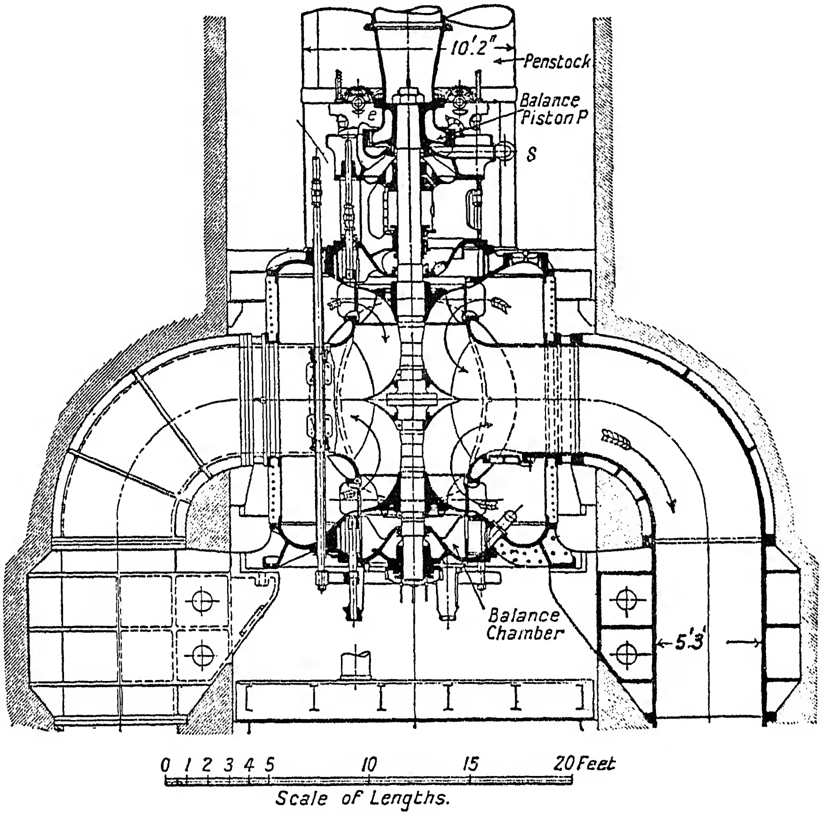

Fig. 12.—Balance Pistons for Vertical Shaft Turbines

necessary. The space C to the left of the balancing disk D is supplied with pressure water from the penstock through a regulating valve, while the space to the right is freely vented. When the disk is in contact with the annular ring S, the force exerted by the pressure water is sufficient to overcome the force acting in the opposite direction on the runner, and the shaft moves to the right. A very small motion allows sufficient leakage between the disk and the seating ring to reduce the pressure in the chamber C to that necessary to produce equilibrium, and in practice the shaft oscillates laterally about this position within very narrow limits. In general the lateral movement does not exceed about ·01 in. Fig. 11 *a* shows one modern design in which the

* Bergstrom, *Proc. Inst. Mech. E.*, February, 1920.

runner has been modified so as to act as its own balancing disk Here the radial clearance between the runner and the casing is reduced to a minimum. The two faced rings on the front and back faces of the runner work, with a normal clearance of about 0·02 in., over corresponding rings on the turbine casing. Any unbalanced force on the runner, say to the left, causes it to move to the right, increasing the clearance between the left-hand rings and reducing that between the right-hand rings. This reduces the pressure in the left-hand space, and increases that in the right-hand space until a position of equilibrium is attained.

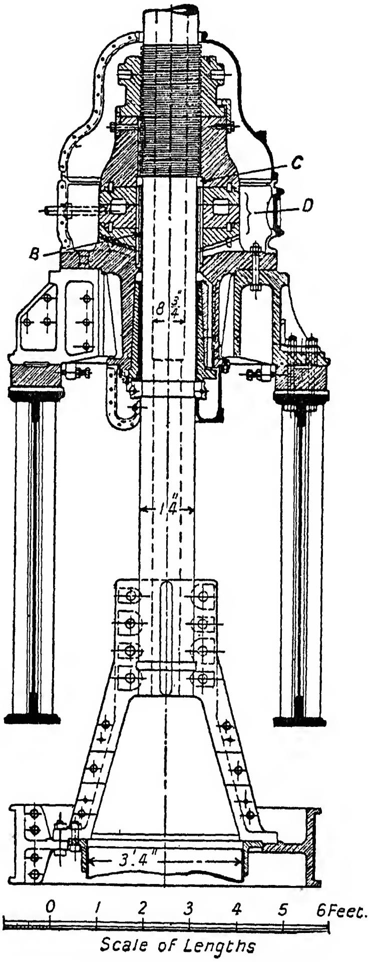

Fig. 13.—Suspension Bearing

Balance Piston for Vertical Turbines.—In the case of a vertical shaft turbine, installed at the bottom of a wheel pit and transmitting its power through a long vertical shaft, the vertical load to be carried is very large, and to obviate the necessity of carrying the whole of this load on a mechanical thrust bearing, balancing pistons have been used in a number of cases. Fig. 12 shows this method as applied in the case of the double Francis turbines of the Canadian Niagara Power Company. Here the weight of the rotating parts is 120 tons, and is balanced partly by the upward pressure on the bottom face of the lower runner, water under the full pressure of the supply head, 133 ft., being admitted to a balance chamber beneath this runner, and partly by the upward pressure on the rotating balance piston, which is subject to the same pressure. Leakage past this piston is drained away to the tail race, and by adjusting the valve on the supply pipe S the upward pressure may be regulated with great nicety.

Any unbalanced load is supported by the suspension bearing (fig. 13) which is placed on the upper deck. In this bearing, oil under a pressure of 375 lb. per square inch is supplied to the annular chamber C surrounding the bush B, and escapes outwards between the fixed and rotating disks at D. These disks have an outside diameter of 36 in. and a bearing area of 780 sq. in. Any slight swing or lateral wear of the shaft is permitted by the spherical bearing of the lower disk.

Thrust Bearings.—The thrust bearing for a horizontal shaft unit is usually of the simple collar type, incorporated as part of one of the main

bearings. Where the end thrust is comparatively small, ring oiled self-lubricating bearings or ball thrust bearings are satisfactory.

Bearings for Vertical Shafts.—In a small unit, erected in an open forebay, the thrust bearing often consists of a plain footstep bearing supporting the lower end of the shaft, and provided with a lignum vitae bearing pad and bearing strips. In large units the bearing is never submerged. It may be carried by a cast-iron truss, supported from the speed ring of the turbine, and situated below the generator, as shown in fig. 7, which illustrates one of the turbines of the Keokuk development on the Mississippi River, or may be situated above the generator on a supporting truss which forms the head cover of the generator, as shown in fig. 8, which shows a more recent development at the Cedars Rapids Power Company. The latter method has the advantage of being more accessible, and gives a more compact construction. The upper guide bearing of the generator is located immediately below the thrust bearing.

Owing to the fact that the whole of the weight of the rotating parts of both turbine and generator is carried by the thrust bearing, this forms a very important feature in the design of such a unit. Until comparatively recently the type of oil-pressure bearing shown in fig. 13 was used almost exclusively for large units. The danger accompanying any failure of the oil supply to such a bearing has led to the more general use, in modern units, of some form of bearing not requiring a pressure oil-supply. By replacing the lower disk by a series of blocks, each of which has an alternate inclined and flat sector, the oil, supplied under gravity, is fed into the segments by the rotation of the shaft, and a simple bearing is produced which is capable of good results under large loads.

An elaboration of this idea has led to the so-called Michell or Kingsbury bearing, in which the stationary disk is made up of several babbitted segments, each of which is mounted on a pivot to enable it to adjust its angle of inclination to the rotating disk, so as to ensure an approximately even distribution of loading over its surface. This bearing also has the advantage that the oil-supply does not need to be under pressure. In such bearings, applied to the Keokuk turbines (fig. 7), the diameter is 56 in., the total load 255 tons, and the mean bearing pressure 350 lb. per square inch. Tests show that the friction loss in these bearings amounts to between 7·5 and 10 kw. at the normal speed of 100 r.p.m., or approximately one-tenth of 1 per cent of the output of the turbine.

In some more recent installations a roller bearing of reduced dimensions is placed inside the main bearing. The roller bearing is made with a slight clearance, and only comes into operation if any wiping of the shoes of the main bearing causes the bearing plate to settle.

The Draft Tube.—The suction or draft tube was originally designed with a view of enabling the turbine to be placed at a convenient height above tail-water level without loss of head. The maximum practical elevation depends on the diameter of the draft tube as indicated in the following table, which is adapted from values given by Meissner:

Diameter of tube, feet	1·0	1·5	2·0	2·5	3·0	4·0	5·0	6·0	8·0
Maximum elevation, feet	30·0	28·0	26·5	25·0	23·5	21·0	19·0	18·0	14·0

From these values should be subtracted the head equivalent to the velocity of flow down the tube.

The draft tube also, if well designed, enables a large proportion of the kinetic energy of discharge from the runner to be converted into pressure

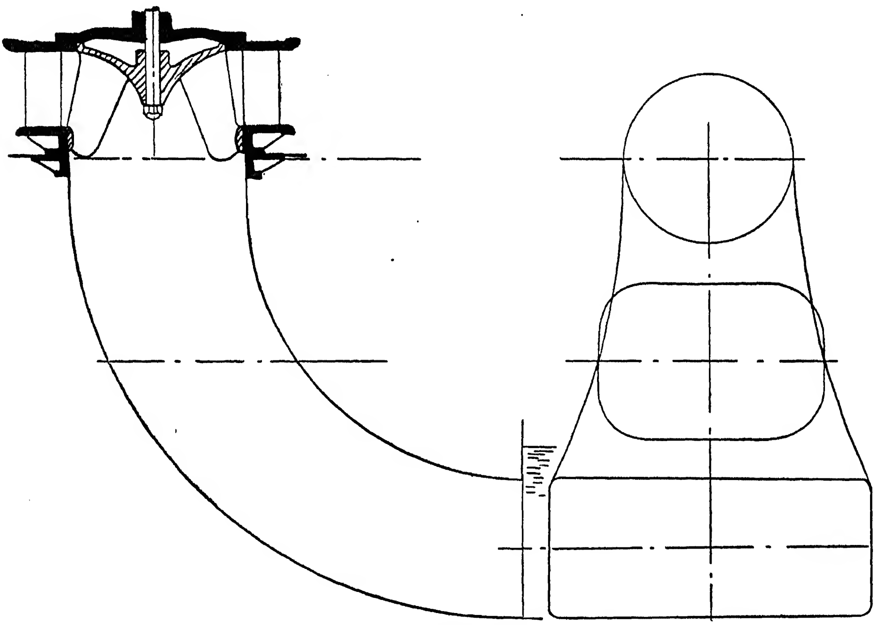

Fig. 14.—Draft Tube

head, and so to be utilized. The mean velocity of discharge from the runner varies from about $0{\cdot}2\sqrt{2gH}$ in a slow-speed turbine under high head to $0{\cdot}5\sqrt{2gH}$, or even more, in a high-speed turbine under a low head. Adopting the latter value, the kinetic energy of discharge represents 25 per cent of the total head, and for high efficiencies in such an installation it is essential that a large proportion of this energy should be recovered. In fact, the high efficiencies of modern low-head plants have only been rendered possible by the most careful attention to this part of the installation.

With a parallel draft tube discharging vertically into the tail race, the whole of the kinetic energy is lost. If, however, the tube is designed with a gradually increasing diameter, so that the velocity is gradually reduced from v_1 to v_2 before discharge, the loss of energy is reduced. The velocity of discharge from the draft tube should not exceed about 4 ft. per second.

The diameter of the draft tube at entrance should be the same as that of the runner to avoid all shock at this point. By curving the tube so as to discharge in the direction of flow in the tail race, the kinetic energy of its discharge is partially utilized in assisting this flow, and is not entirely wasted.

The draft tube may be either of steel-plate or cast-iron construction, or may be moulded in the concrete substructure. The latter arrangement, if not ruled out on the ground of expense, is preferable, since it enables any desirable form of cross section to be adopted, whereas with a steel-plate construction the conical circular form is almost essential. With a moulded draft tube, a form such as fig. 14, which enables a comparatively shallow tail race to be used, presents no difficulties.

With a horizontal shaft unit, the turbine shaft passes through the draft tube, and a stuffing box is necessary to prevent air leakage into the tube. Air-tightness may be assured by means of a water seal consisting of a chamber surrounding the shaft, and supplied with pressure water from the penstock through a mall pipe.

Hydraulics of the Reaction Turbine.—The space available renders it impossible to do more than touch on the general principles on which the hydraulic design of the reaction turbine is based.*

In the following discussion let

ω = angular velocity of the runner in radians per second,
$\omega = 2\pi N \div 60$ where N = revolutions per minute,
$u = \omega r$ = velocity of wheel at point indicated by a suffix,
v = absolute velocity of water,
w = tangential component of v,
f = radial component of v,
v_r = relative velocity of water and vane,
α = guide vane angle (fig. 15),
β = wheel vane angle at entrance,
γ = wheel vane angle at exit,
Q = flow in cubic feet per second,
W = weight of 1 c. ft. of water,
suffix (2) refer to inlet to wheel vanes,
,, (3) refer to exit from wheel vanes.

For entry without shock, the direction of the relative velocity of water and vane at entrance to the wheel is to be parallel to the vane tips, and a consideration of the diagram of velocities (fig. 15) shows that if the angles are correctly proportioned:

$$f_2 = w_2 \tan a = (w_2 - u_2) \tan\beta; \therefore u_2 = w_2\left(1 - \frac{\tan\alpha}{\tan\beta}\right).$$

$$f_3 = (u_3 - w_3) \tan\gamma.$$

$$_2v_r = f_2 \operatorname{cosec}\beta; \ _3v_r = f_3 \operatorname{cosec}\gamma.$$

* For further information the reader is referred to some such book as that of Gelpke and van Cleeve, *Turbines and Turbine Installations*.

$$\left.\begin{array}{r}\text{The change of the moment of momentum}\\ \text{in the wheel} = \text{turning moment}\end{array}\right\} = \frac{WQ}{g}\{w_2r_2 - w_3r_3\} \text{ ft. lb.}$$

$$\therefore \left.\begin{array}{r}\text{Work done per second}\\ \text{on runner}\end{array}\right\} = \frac{WQ}{g}\{w_2r_2 - w_3r_3\}\omega$$

$$= \frac{WQ}{g}\{w_2u_2 - w_3u_3\} \text{ ft. lb.}$$

In an ideal wheel, with no friction or eddy losses, we should have, neglecting changes of level in the wheel,

$$\frac{p_2}{W} + \frac{v_2^2}{2g} = \frac{p_3}{W} + \frac{v_3^2}{2g} + \text{work done per pound between (2) and (3).}$$

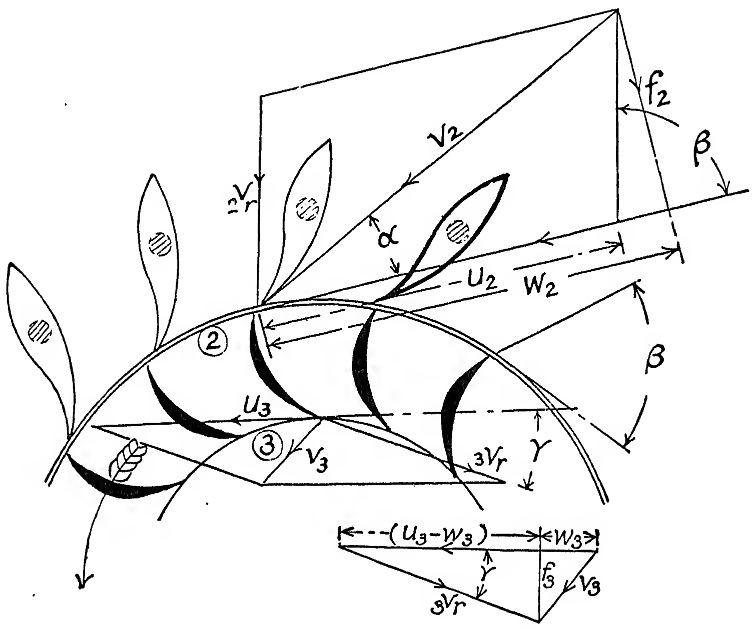

Fig. 15

The efficiency will be a maximum when the energy rejected in the discharge is a minimum, i.e. when v_3 is a minimum, or when w_3 is zero, in which case $v_3 = f_3$. Assuming the wheel to be designed for this state of affairs,

$$\frac{p_2}{W} + \frac{v_2^2}{2g} = \frac{p_3}{W} + \frac{f_3^2}{2g} + \frac{w_2u_2}{g}.$$

Writing H as the head available to produce flow through the wheel, so that

$$H = \frac{p_2}{W} + \frac{v_2^2}{2g} - \frac{p_3}{W}, \text{ we have } H = \frac{f_3^2}{2g} + \frac{w_2u_2}{g},$$

from which, writing $f_3 = f_2 \dfrac{b_2r_2}{b_3r_3} = w_2 \tan\alpha \dfrac{b_2r_2}{b_3r_3}$,

where b and r are the breadth and radius of the wheel, we get, on substitution,

$$w_2 = \sqrt{\frac{2gH}{2 + \left(\frac{b_2 r_2}{b_3 r_3}\tan\alpha\right)^2 - 2\,\frac{\tan\alpha}{\tan\beta}}},$$

$$\text{while } u_2 = w_2\left(1 - \frac{\tan\alpha}{\tan\beta}\right),$$

$$\text{and } Q = A_2 f_2 = A_2 w_2 \tan\alpha.$$

Thus in a wheel of given design, the peripheral speed for maximum efficiency and the volume of discharge each vary as $\sqrt{H}$, while the output of the turbine, being proportional to Qu_2w_2, varies as $H^{\frac{3}{2}}$.

$$\text{The hydraulic efficiency } \eta = \frac{\text{work done per pound}}{H}$$

$$= \frac{w_2 u_2}{gH},$$

$$\text{which on substitution becomes } \frac{1}{1 + \frac{1}{2}\left(\frac{b_2 r_2}{b_3 r_3}\tan\alpha\right)^2\left(\frac{\tan\beta}{\tan\beta - \tan\alpha}\right)}.$$

Writing $u_2 = k\sqrt{2gH}$, the manner in which k and therefore the speed of the wheel, and in which η vary with changes in α and β is shown in the following table for the special case in which $b_2r_2 = b_3r_3$.

Values of β.		Values of α.				
		10°	15°	20°	25°	30°
60°	k	0·658	0·636	0·604	0·564	0·516
	η	0·981	0·959	0·922	0·871	0·800
90°	k	0·702	0·695	0·685	0·672	0·655
	η	0·984	0·964	0·936	0·902	0·857
120°	k	0·741	0·748	0·756	0·764	0·770
	η	0·986	0·968	0·946	0·920	0·889
150°	k	0·802	0·845	0·885	0·924	0·960
	η	0·988	0·971	0·959	0·940	0·917

The results show that by suitable adjustment of α and β, the peripheral speed for a given head may be varied between wide limits. For high speeds β should be large, and for high efficiencies α should be small. As β is increased the value of f_2 and hence the volume of water passing a wheel of given size diminishes, so that to obtain the same output the size of the wheel must be increased. If, as is usually the case in low-head plants, a high rotative

speed is required, the inlet area is increased by increasing the depth of the runner. Such a turbine has a comparatively large ratio of inlet area to discharge area, and the velocities of discharge are relatively high. For high heads β may be between 60° and 90°, and, for medium and low heads, between 90° and 135°.

Similarly, while the hydraulic efficiency decreases as α increases, the volume of flow increases with α, and the maximum output is obtained when the product of Q and η is a maximum. For high efficiency α should be as small as mechanical considerations permit, generally between 12° and 18°. Where a cheap turbine is required and the efficiency is not of great importance, α may have a value as high as 35°.

Specific Speed of a Turbine.—If P denote the output in b.h.p., then, for a given turbine,

(1) the speed N is proportional to $\sqrt{H}$,
(2) the discharge Q is proportional to $\sqrt{H}$,
(3) the output P is proportional to H $\sqrt{H} = H^{\frac{3}{2}}$.

In order to afford a basis of comparison of turbines of different diameters and proportions operating under different heads, the term known as "specific speed" has been introduced. This may be defined as the speed at which a runner would operate if reduced geometrically to such a size that it would develop 1 h.p. under unit working head. The figures for specific speed given in the following pages refer to a unit head of 1 ft. If the metre be adopted as the unit, these figures require to be multiplied by 4·45.

To determine the value of the specific speed, imagine the head to be reduced from H to h, the dimensions remaining unaltered. Then, since the peripheral speed for maximum efficiency is proportional to $\sqrt{H}$, while the output is proportional to H, we have:

$$\frac{n}{N} = \sqrt{\frac{h}{H}}, \text{ while } \frac{p}{P} = \left(\frac{h}{H}\right)^{\frac{3}{2}}.$$

Now imagine all the proportions of the turbine to be reduced in the same ratio. Since for a given head the number of revolutions is proportional to the diameter, and since the quantity of water is proportional to the inlet area, and therefore to the square of the diameter, we have

$$\frac{p}{P} = \left(\frac{h}{H}\right)^{\frac{3}{2}} \times \left(\frac{d}{D}\right)^2$$

$$\begin{aligned} \text{and } \frac{n}{N} &= \sqrt{\frac{h}{H}} \times \frac{D}{d} \\ &= \sqrt{\frac{h}{H}}\sqrt{\frac{P}{p}} \times \left(\frac{h}{H}\right)^{\frac{3}{4}} \\ &= \sqrt{\frac{P}{p}} \cdot \left(\frac{h}{H}\right)^{\frac{5}{4}}. \end{aligned}$$

If now h and p be made equal to unity, n becomes the specific speed N_s, so that

$$N_s = \frac{N\sqrt{P}}{H^{\frac{5}{4}}}.$$

In the case of a turbine having more than one runner, P in this expression represents the output per runner. If P is taken as representing the total output, this value of N_s is to be divided by the square root of the number of runners.

The specific speed of a reaction turbine may be varied by varying the diameter of the runner, the angle of the guide vanes, and the angle of the wheel vanes. By modifying the design as indicated in the sketches of fig. 9 it is possible, while maintaining high efficiencies at full load, to increase the specific speed from about 15, its minimum value with the type shown in fig. 9 A, to about 125 with the type shown in fig. 9 D. Specific speeds as high as 150 are possible with some sacrifice in efficiency, and it is probable that further developments will see the value increased still further. The turbine shown in fig. 3 has a specific speed of 118·5 at its designed speed of 93·7 r.p.m. under a head of 32·8 ft. These high specific speeds are extremely valuable for low-head installations, since they enable the size and cost of the turbine, of its setting, and of the generator to be greatly reduced. In fact, many existing low-head installations would have been commercially impracticable but for the development during recent years of the high-speed turbine.

High specific speeds are, however, attended by some disadvantages, particularly for medium and high heads. The part gate efficiency in general falls off as the specific speed increases. Also if the speed is unduly high it becomes very difficult to avoid corrosion troubles. At the present stage of design, the maximum specific speeds to be used under normal circumstances with various heads are approximately as follows:

Head (feet) ..	20	40	60	80	100	150	200	300	400	600
Specific Speed..	125	100	85	70	65	50	43	33	30	25

In the earlier low-head plants, sufficiently high speeds were attained by the use of two or more runners on the same shaft, as shown in fig. 16. This arrangement has a number of disadvantages as compared with the single runner. These are:

1. A separate series of guide vanes or gate mechanisms is required for each runner, one or all of which are submerged.

2. Torsional deflection in the operating shafts renders it difficult to ensure equal gate openings on all runners.

3. Owing to shock and interference between the discharge streams from the runners of a double-runner turbine, a greater proportion of the kinetic

energy of discharge is lost than in the case of a single-runner turbine. This effect is increased by the impossibility of avoiding sudden changes of direction of flow in the water leaving the casing of the double turbine.

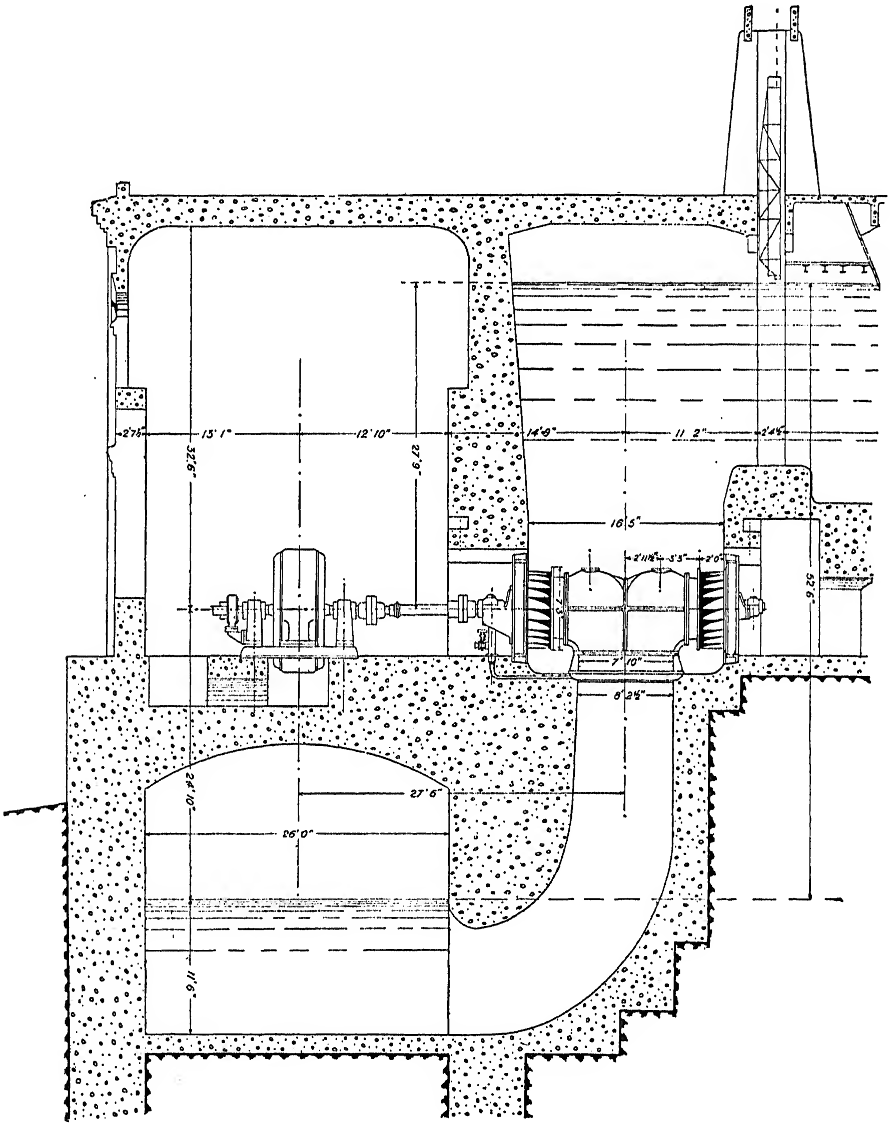

Fig. 16.—Twin-runner Open-type Turbine, with Back Bearing in Inspection Tunnel

With a single-runner unit, on the other hand, only one gate mechanism is necessary; this is outside the turbine casing and accessible for inspection and repair at all times; and it becomes possible to adopt the type of moulded volute construction shown in fig. 17. Development, especially in the United States, has of recent years been tending to a more general use of this vertical

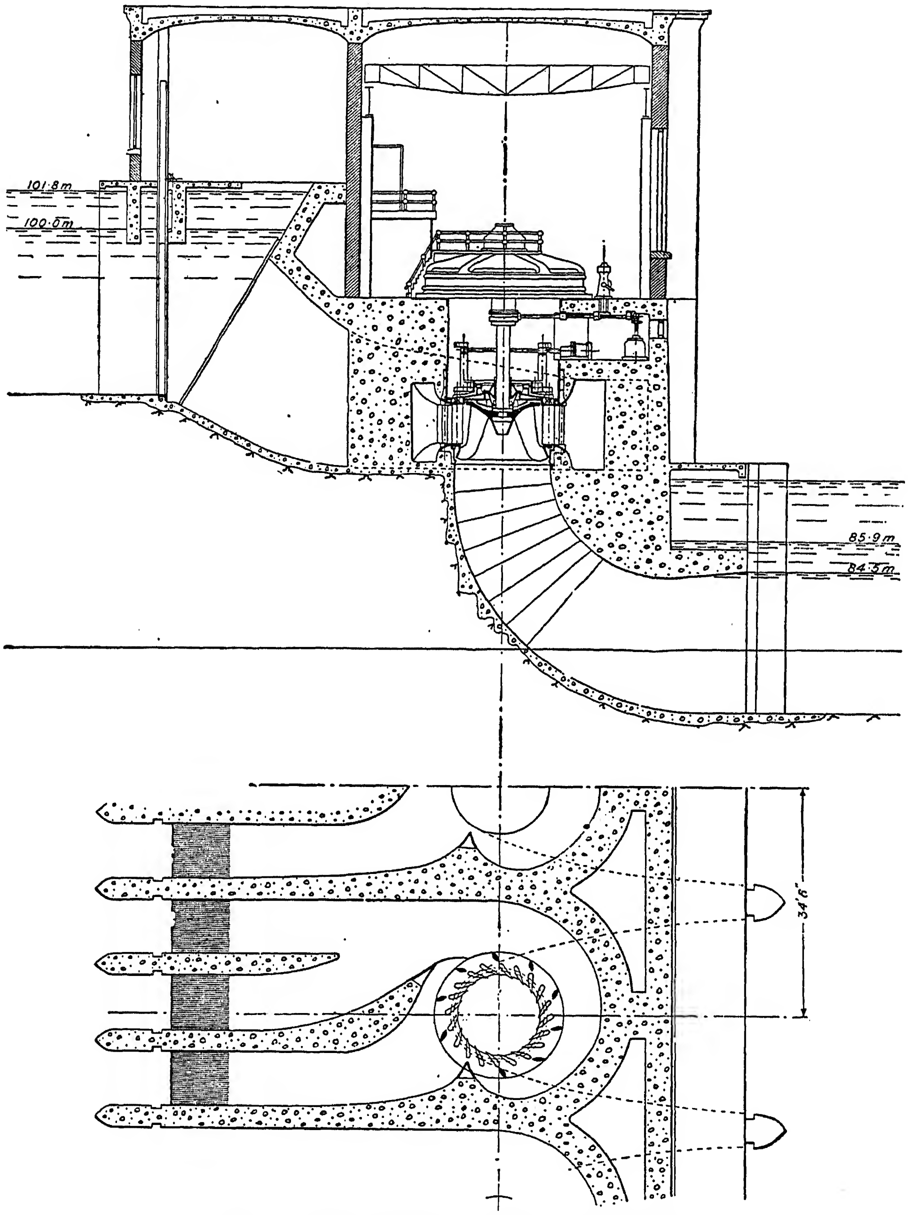

Fig. 17.—Single-wheel Direct-coupled Vertical Shaft Units

single-runner type of unit for medium and even high heads, as well as for low-head plants, except where the general arrangements call for an overhead intake to the turbines.

Nominal Diameter of a Turbine Runner.—In a low-speed reaction turbine of the types shown in fig. 9 A and B, the vane edges at the point of entry are parallel to the axis of the shaft. The diameter is measured at

this point, and has a perfectly definite value. In a high- or medium-speed wheel the inlet edges of the vanes are usually inclined, while the discharge edge of the bucket has a maximum diameter which is greater than any point on the inlet side. In this case there is no definite convention as to what

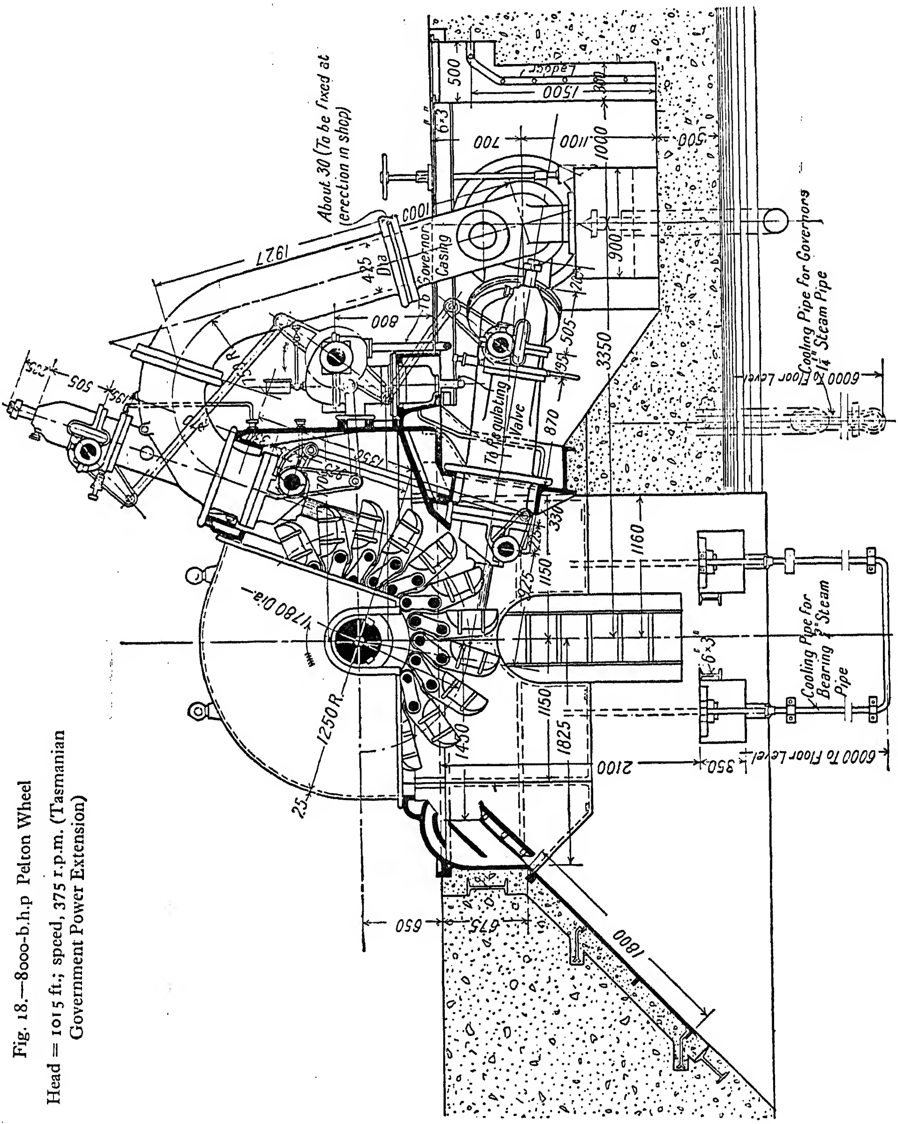

Fig. 18.—8000-b.h.p Pelton Wheel

Head = 1015 ft.; speed, 375 r.p.m. (Tasmanian Government Power Extension)

point shall be taken as giving the diameter of the wheel, and this discrepancy accounts to some extent for the fact that a turbine given as a certain diameter by one maker will have a greater capacity, speed, or output than one nominally of the same diameter quoted by another maker.

In a turbine of either of types 9 C or 9 D, the maximum diameter,

measured over the discharge edge of the buckets, is most commonly given. In any estimate involving the diameter of the runner, the prospective buyer should satisfy himself at which point this is measured.

The Pelton Wheel.—Except for very large units the Pelton wheel

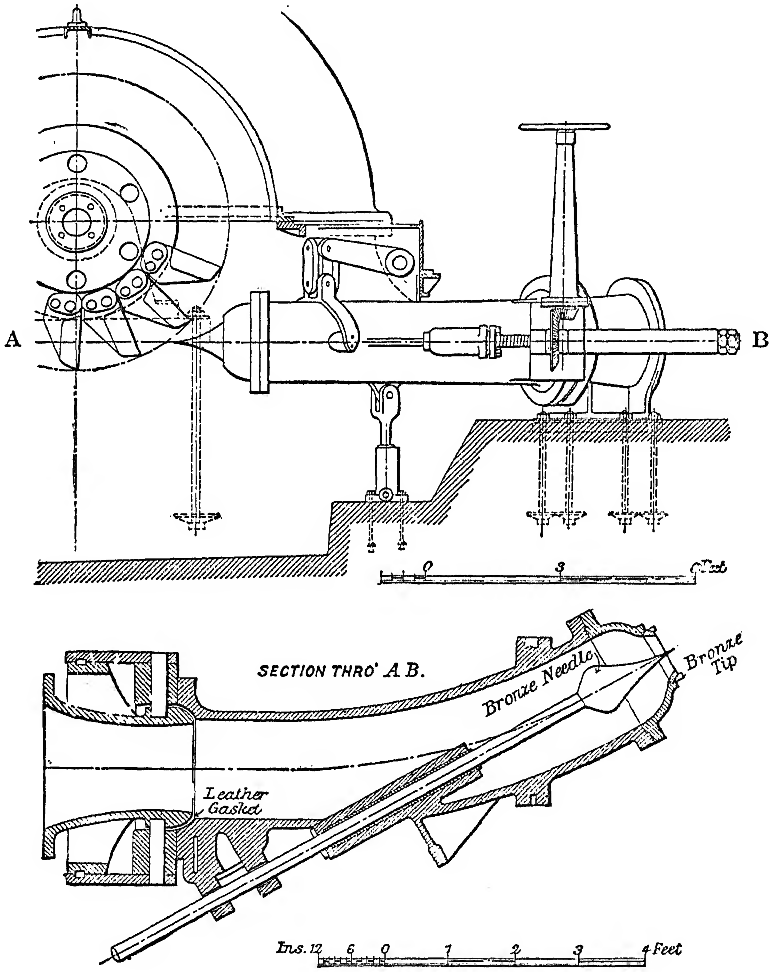

Fig. 19.—Deflecting Nozzle with Hand Adjustment to Needle

is seldom constructed with a vertical shaft, owing to the advantages of simplicity of construction and facility for inspection afforded by the horizontal shaft type. In general only one nozzle is used on a single wheel. Two jets have been used in a number of cases, and in this way the power can be practically doubled. Such jets are usually placed approximately at right angles, as shown in fig. 18.* It is found, however, that in a horizontal shaft unit the splash from one jet affects the other, and reduces the

* *Engineering*, 2nd July, 1920, Messrs. Boving & Co., Ltd., London.

efficiency somewhat. Where the output required from a unit is greater than can be obtained from a single jet, it is usually preferable to mount two single-jet wheels side by side on the same shaft.

Nozzles.—The modern Pelton wheel is always fitted with a circular nozzle, with an axial needle or spear for regulating the size of the jet. Other shapes of jet have been used, but all such forms suffer a greater windage loss than the circular. Also all other forms tend to become circular, and in the process tend to become unsteady. The maximum practicable diameter of jet appears to be about 12 in. The axial position of the needle in the nozzle is regulated either by hand (fig. 19), or, in all important installations, by the governing mechanism, as in fig. 18. For high efficiencies the diameter of the pitch circle of the buckets should not be less than about twelve times the diameter of the jet.

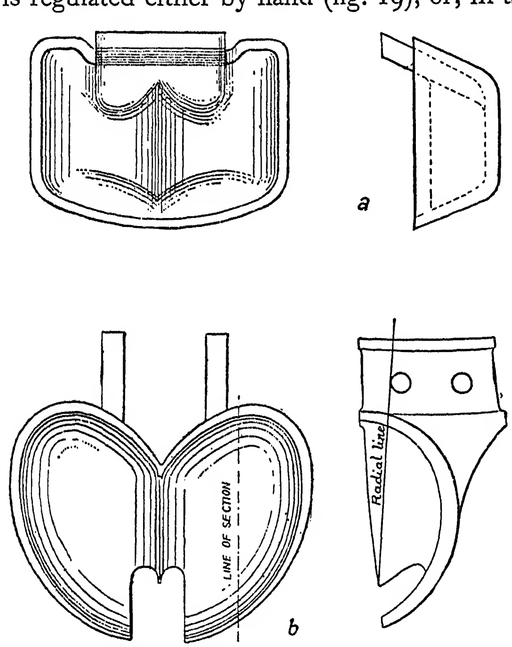

Fig. 20.—Types of Pelton Buckets

Buckets. — The original Pelton buckets were of rectangular section (fig. 20 *a*). These have been superseded by the elliptical bucket (fig. 20 *b*), in which that part of the lip in the line of the jet is omitted. The lip and ridge of the original bucket deflect the jet in two planes at right angles, and as the paths of the streams thus formed cross, a certain amount of energy is dissipated by their impact. Also the lip tends to deflect the jet radially inwards towards the rim of the wheel, in which case some fouling of the succeeding bucket is inevitable. The sharp curves and corners of this type of bucket cause an appreciable loss in eddy formation, and tests show that the efficiency obtained with the modern form of bucket is from 6 to 10 per cent greater than with the older form.

The angle through which the jet is deflected by the bucket should be as nearly 180° as possible. In order that the discharge from one shall clear the back of the following bucket, in practice this angle is limited to a maximum of about 165°.

The friction loss in the buckets increases with the wetted area, and to reduce this the number of buckets should be as small as is consistent with continuous impact, while they should be made no larger than is necessary to give the required change of direction with easy curves and without shock. The surface should be as smooth and well finished as possible. In modern

practice the width of the buckets is between three and four times the jet diameter, the ratio diminishing as the size of jet is increased. In a high-speed wheel the runner consists of a steel disk, to which the buckets are bolted. The latter usually carry two lugs which straddle the wheel rim, and are attached to it by two bolts. In some cases a double-wheel disk has been used with the buckets carrying three lugs. Two of these straddle the

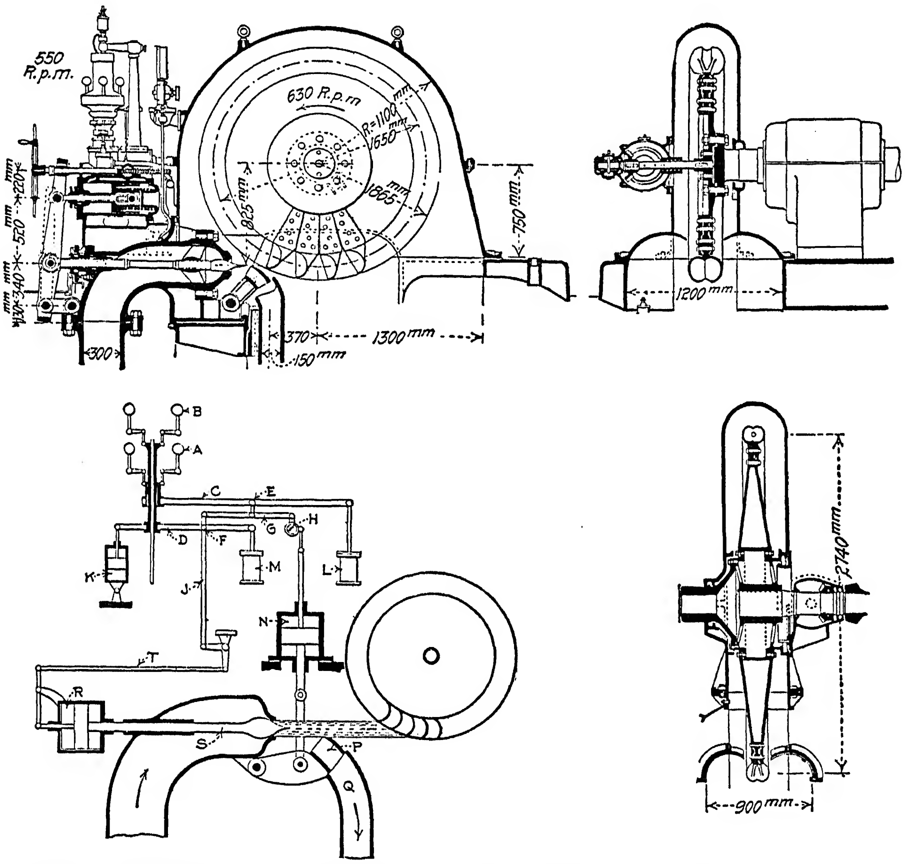

Fig. 21.—Arniberg Power Station, 3000 h.p. Head = 2800 ft., 630 r.p.m. Pitch circle, diameter = 5·41 ft.

disks, and the third fits between the disks. The lugs are so designed that a single bolt passes through the rear lug of one bucket, and the forward lugs of the next. This type of construction is advantageous where large buckets are to be fitted to a wheel of comparatively small diameter.

Fig. 21 shows a type of construction adopted at the Arniberg Power Station. Here units developing respectively 3000 h.p. and 1300 h.p. under 2800-ft. head have wheels of about 5 ft. 5 in. and 9 ft. 6 in. diameter, and make respectively 630 r.p.m. and 360 r.p.m. In both these wheels rivets and not bolts are used, so that individual buckets cannot be replaced.

Hydraulics of the Pelton Wheel.—If H be the pressure head behind

the nozzle of a Pelton wheel, the velocity of efflux is equal to $C_v\sqrt{2gH}$ ft. per second, where C_v, the coefficient of velocity, in a well-formed needle nozzle is approximately 0·99. Writing V_1 for $\sqrt{2gH}$, the horse-power of the jet is equal to

$$\frac{62{\cdot}4aC_v{}^3V_1{}^3}{550 \times 2g},$$

where a is the area of the jet in square feet. Giving C_v the value 0·99, this expression equals $0{\cdot}00171aV_1{}^3$ h.p.

Let u = peripheral speed of buckets at pitch circle.
,, V_2 = final absolute velocity of water leaving the buckets.
,, ${}_1v_r$ = relative velocity of jet and bucket at entrance.
,, ${}_2v_r$ = relative velocity of jet and bucket at discharge.
,, α = mean angle between jet and tangent at point of contact.
,, γ = total angle of deflection of jet. (Fig. 22.)

Then the initial velocity of jet in direction of tangent at point of impact $= V_1 \cos\alpha$.

The component, parallel to the tangent at discharge, of final velocity relative to bucket $= {}_2v_r \cos\gamma$.

∴ Absolute velocity in this direction at discharge $= u + {}_2v_r \cos\gamma$.

∴ Change of tangential momentum per second per pound

$$= \frac{1}{g}\{V_1 \cos\alpha - u - {}_2v_r \cos\gamma\}.$$

∴ Work done per pound of water per second

$$= \frac{u}{g}\{V_1 \cos\alpha - u - {}_2v_r \cos\gamma\} \text{ ft. lb.}$$

$$\therefore \text{Efficiency} = \frac{u}{gH}\{V_1 \cos\alpha - u - {}_2v_r \cos\gamma\}.$$

The loss due to friction and eddies in the buckets $= \dfrac{{}_1v_r{}^2 - {}_2v_r{}^2}{2g}$ ft. lb. per pound, where ${}_1v_r = \sqrt{V_1{}^2 + u^2 - 2V_1u \cos\alpha}$.

The loss due to rejection of kinetic energy in the discharge $= \dfrac{V_2{}^2}{2g}$ ft. lb. per pound, where $V_2{}^2 = u^2 + {}_2v_r{}^2 + 2u_2v_r \cos\gamma$.

Tests show that in an average wheel ${}_2v_r$ may be as low as from 0·5 to 0·6 ${}_1v_r$. In a well-designed bucket, however, having a ratio of bucket width to jet diameter not less than about 3·3, this ratio approximates to 0·75 or even 0·8. If the angle of deflection were 180°, and if the buckets were frictionless, the value of the peripheral speed of the wheel for maximum efficiency would be $V_1 \cos\alpha \div 2$, or approximately $V_1 \div 2$, since α is small. When account is taken of the loss in the buckets and of the fact that γ is less than 180°, the

best peripheral speed lies between 0·44 and 0·48 V_1, the higher ratio being possible with the most efficient buckets.

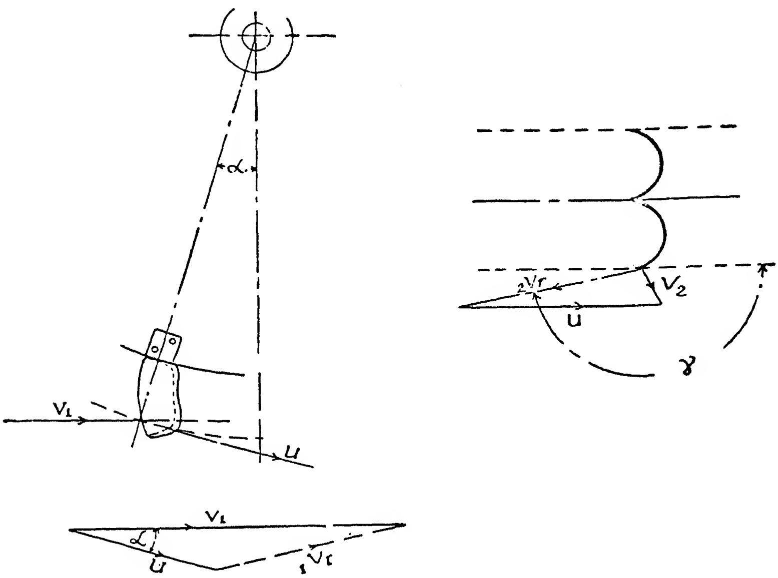

Fig. 22

Specific Speed.—As for a reaction turbine, the specific speed of a Pelton wheel is given by

$$N_s = \frac{N\sqrt{P}}{H^{\frac{5}{4}}}.$$

But $N = \dfrac{60u}{\pi D}$, where D is the diameter of the pitch circle,

and $P = \dfrac{0{\cdot}00176\pi d^2}{4} V_1{}^3\eta$, where d is the diameter of the jet

$$= \frac{0{\cdot}00176\pi d^2}{4}\,\eta(2gH)^{\frac{3}{2}}$$

$$= 0{\cdot}610 d^2 H^{\frac{3}{2}} \text{ if the efficiency } \eta = 0{\cdot}85.$$

$$\therefore \sqrt{P} = 0{\cdot}780 d H^{\frac{3}{4}}.$$

$$\text{Also } u = 0{\cdot}46\sqrt{2gH} \text{ (approx.)}.$$

$$\therefore N_s = \frac{60 \times 0{\cdot}46\sqrt{2gH} \times 0{\cdot}780 d H^{\frac{3}{4}}}{\pi D H^{\frac{5}{4}}}$$

$$= \frac{55d}{D}.$$

If $D \div d = 12$, this makes the specific speed approximately 4·6. This is the highest value of the specific speed for a single-jet Pelton wheel for high

efficiency. With two jets, or two wheels each with a single jet, the maximum value with the same ratio of D : *d* becomes approximately 6·5. By reducing the ratio D : *d* to 9, the specific speed, with a single jet, becomes approximately 6·0. The full load efficiency of such a wheel is about 5 per cent less than that of a wheel with a specific speed of 5·0, and in order to obtain this efficiency the buckets require to be very carefully designed.

Speed Regulation of Pelton Wheels.—There are three general methods of regulating the speed of Pelton wheels. These are:

1. By deflecting the jet from the wheel by a deflecting nozzle or hood.
2. By a combined needle regulator and pressure regulator.
3. By a combined needle regulator and deflector, or deflecting nozzle.

In the first method the jet is deflected wholly or partly from the wheel by the action of a relay piston operated from the governor, or in small plants by the governor directly. The flow in the pipe line is independent of the load, no pressure surges are produced, and the method gives excellent governing. It is, however, very wasteful under a variable load, and is only to be condoned where, as in some irrigation schemes, or in some installations on the higher reaches of a stream, the user is under an obligation to discharge water at a minimum rate for users lower down the stream. The loss may be reduced by using the deflector in conjunction with a hand-regulated needle. This is set by hand at intervals to give the maximum discharge likely to be required during the next period, and any fluctuation of load up to this maximum is handled by the deflector.

In the earlier forms of deflecting nozzle (fig. 19) * the nozzle is connected to the pipe line through a swivel joint packed by means of a leather ring, and is carried by the plunger rod of the relay cylinder. The pressure below this plunger and hence its position are regulated by a pilot valve operated by the governor.

Fig. 23 † shows a deflecting nozzle in which the nozzle with the needle-regulating mechanism is mounted on a hollow trunnion through which the pressure water is conveyed. This is carried through a stuffing box which is easily kept tight. Instead of a deflecting nozzle, the jet may be deflected by means of a hood or deflector, fitted between the nozzle and the wheel, and whose position is regulated by the governor (fig. 21). This is lighter, cheaper, and gives less trouble in maintenance than the deflecting nozzle.

In the second method of regulation, the position of the needle is regulated by the governor, while to prevent pressure surges accompanying a sudden reduction of load a pressure regulator or relief valve is fitted. While, owing to the smaller volumes of water to be handled, the pressure regulator is more suitable for Pelton wheels than for reaction turbines, it suffers from the liability of the valve to stick, and from the difficulty of ensuring its synchronous action, and in most recent plants the third method of regulation has been used.

* *Proc. Inst. Mech. Engineers*, January, 1910. † *Engineering*, 28th August, 1914.

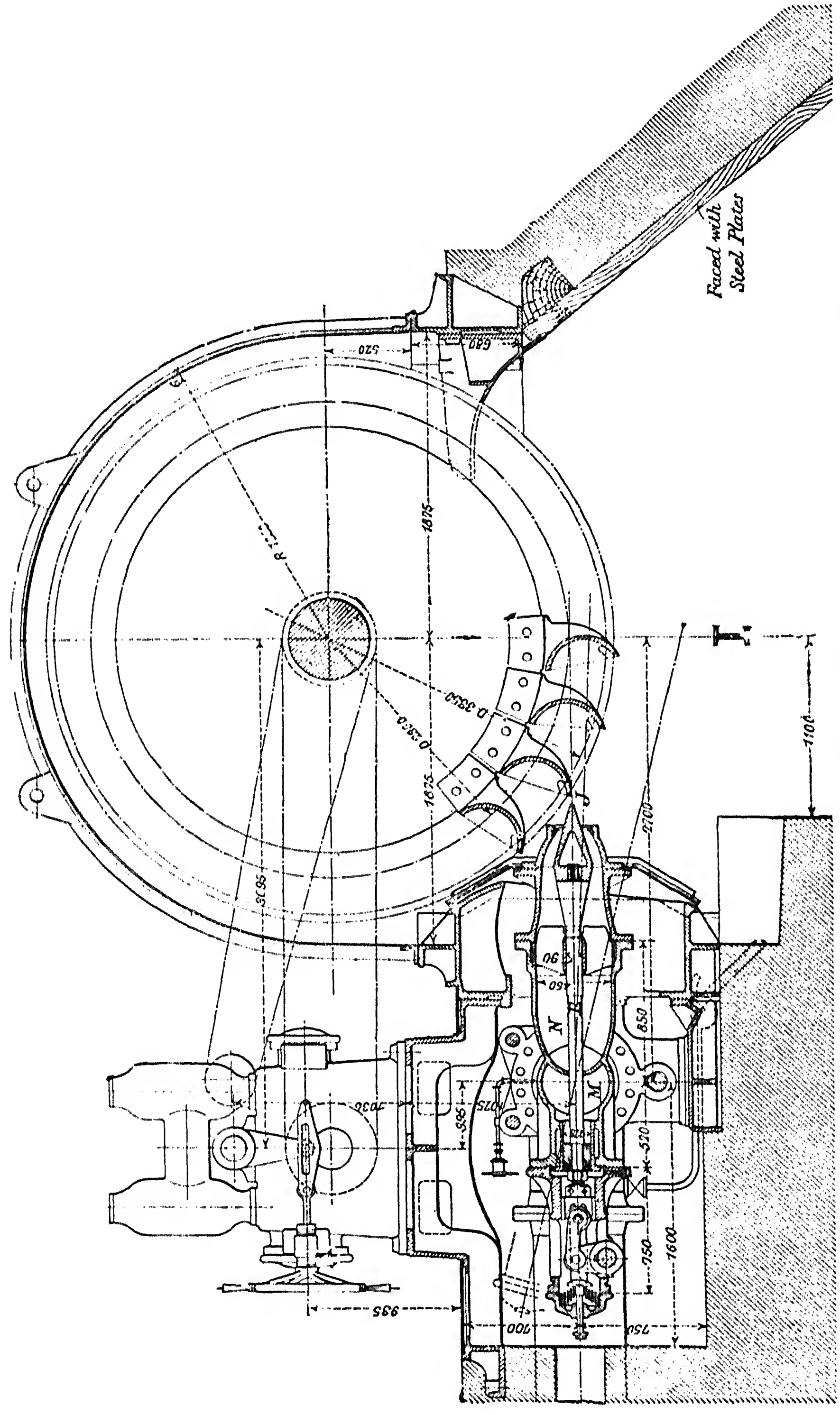

Fig. 33.—13,000-h.p. Pelton Wheel at Tyssedal Power Station. Head = 1330 ft., 250 r.p.m. Pitch circle, diameter = 9·52 ft.

Here both deflector and needle are regulated by the governor. The connection is so arranged as to allow of a rapid movement of the deflector on a reduction of load, followed by a slow movement of the needle closing the nozzle. When in its final position, the deflector is tangential to the reduced jet, ready for an immediate response to any fresh change of load. One arrangement of this kind is shown in fig. 24.* The needle carries a slotted extension G, and is forced to the right by a spring H, whose action is retarded by the action of the oil dashpot C. If the load is reduced the lever F is forced to the right by the servo-motor, deflecting the jet, while the needle follows slowly until the slot is again in contact with the end of the lever F, with the needle tangential to the jet. On an increase in load the lever is forced to the left, and carries with it both the needle and deflector. The valves J in the dashpot plunger allow free motion in this direction.

Fig. 21 shows another method of obtaining the same result, as applied to the units of the Arniberg Power Station. Here the governor has two flyball systems acting on two levers A and B, with fulcrums at E and F. The fulcrum E is carried by the bar G, which has one end connected to the eccentric H, and the other to a rod J. The lever D is connected to a dashpot K, and the levers C and D control the regulating valves L and M respectively. The valve M controls the piston of the servo-motor R, which controls the needle S. The valve L controls the piston N and the deflector P. For a slow change of speed, levers C and D move in unison. The needle and the deflector are moved simultaneously, the proportions being so fixed that the deflector is brought finally into a position tangential to the jet. If the change of load is rapid, however, the dashpot prevents the free motion of the lever D. In consequence the deflector acts as before, but the action of the jet is much slower. When the size of the jet has been reduced to that corresponding to the reduced load, the deflector, through the agency of the rods T and J and the lever G with the eccentric H, is again brought tangential to the jet. The pipe line is 6850 ft. long, and varies in diameter from 18 to 24 in. Tests show that if 97 per cent of the full load is suddenly thrown on, the initial diminution of pressure is 11 per cent and the speed variation 17 per cent. When 97 per cent of the full load is suddenly thrown off, the initial increase of pressure is 5 per cent and the speed variation 5·5 per cent.

Runaway Speeds.—If, when load is thrown off the turbine, a failure of the governing mechanism or jamming of the gates causes the latter to remain open, the speed will increase considerably, and the rotors of the turbine and of the electric generators should be designed so as to be safe under the maximum runaway speeds which may be attained.

In a Pelton wheel installation, the maximum possible peripheral speed will be somewhat less than that of the jet owing to mechanical friction and windage, and as the normal speed is slightly less than one-half that of the jet, the runaway speed should be taken as twice the normal speed.

* By courtesy of Messrs. The English Electric Company, Ltd., London.

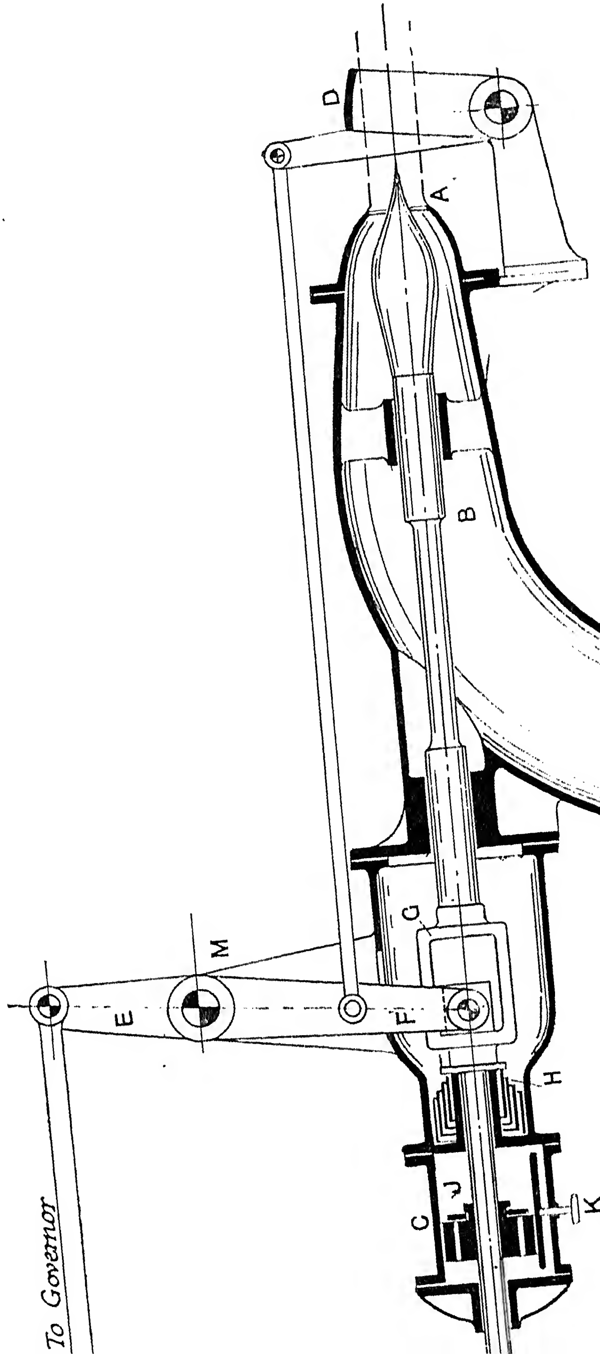

Fig. 24.—Pelton Nozzle with Governor Regulation to Deflector and Needle

With reaction wheels working under a constant head, and operating normally at the most efficient speed, the runaway speed is between 55 and 85 per cent above the normal speed. In a low-head turbine operating under a wide variation in the head, and designed for maximum efficiency under the average head, the runaway speed under the maximum head may be as much as three times the normal operating speed. This depends essentially on the range of head and on the design of the turbine, and in such an installation the runaway possibilities should be carefully investigated, with reference to actual tests on a wheel of similar design.

Several types of over-speed regulators are available. These usually consist of a separate centrifugal governor operating an emergency control which, when coming into operation, shuts down the unit.

Selection of Turbine.—The choice of the most suitable type of turbine depends upon a number of factors, including the power, the head, the desired speed of rotation, and the special circumstances of the plant.

If the head is so large that its variations under varying conditions of seasonal flow can be neglected, the turbine can be considered from the view of constant head operation. If the head is low, the rise in level of the tail water in times of flood is usually much greater than that of the head water, and the percentage variation in head may become large. Under such conditions a turbine should be selected which will give satisfactory operation and reasonable efficiencies over the entire range of heads, and which will give high efficiencies under the highest heads, which occur when the minimum quantity is available. Some of the recent high-speed runners are extremely well adapted for such service.

Where the load variations are likely to be large and where the quantity of water is limited, a turbine should be selected which will give its maximum efficiency under the average conditions of operation.

It is usually necessary to select a turbine having a given speed and output in order to operate a generator for which these characteristics are fixed.

From a knowledge of the necessary output and speed, the characteristic speed is deduced from the expression

$$N_s = \frac{N\sqrt{P}}{H^{\frac{5}{4}}}.$$

Having determined the characteristic speed, a type of turbine should be selected having a value of N_s, not less than the desired value, and, for high efficiency, not greatly different from this value. If the calculated value for a single turbine is greater than is attainable with the type selected, the power must be divided between two or more units, until the required conditions are satisfied.

If, for example, an output of 20,000 h.p. be required at 150 r.p.m. under a head of 40 ft., the value of N_s for a single wheel would be 211. As this is higher than is attainable with any of the types in general use, it would be necessary to install more than one unit. The following table shows the

number of turbines or of runners which would be necessary, with the corresponding specific speeds:

Number of runners ..	3	4	5	6	8	10
Specific speed ..	122	106	94	86	75	67

The question as to whether a vertical- or horizontal-shaft machine, or one having a single runner or two or more runners is most suitable depends largely on the special circumstances of the installation. The all-important factors guiding the decision should be the reliability, the accessibility, and the efficiency of the unit. In the average hydro-electric installation the cost of the turbine itself and of its setting forms such a very small proportion of the total cost of the scheme, seldom exceeding 5 per cent of the total, that no question of its cost should be allowed to weigh against the advantages of freedom from breakdown, ease of repair, and efficiency of operation. It is to be remembered that any increase in efficiency enables the capacity of the head works, the head race, and the pipe line to be correspondingly reduced, and, with a limited quantity of water, enables just so much more electrical energy to be developed and sold. Generally speaking, the saving in these respects far outweighs any additional costs incurred by installing the most efficient turbine which can be obtained.

REFRIGERATION

BY

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Refrigeration

Introduction

Practically all refrigeration, other than that produced by the melting of natural ice, is now obtained by the evaporation of a volatile liquid. The vapour so produced is condensed back to a liquid which in turn is re-evaporated, producing more cold, the process being a continuous one. The whole science of mechanical refrigeration is in fact governed by the following simple physical laws:

1. Liquids, when evaporating, abstract heat from their surroundings.
2. The temperature at which a liquid evaporates depends on the pressure to which it is subjected.
3. All vapours can be condensed to liquids by being suitably compressed and cooled.

The first law explains why artificial cold can be produced; the second law explains how different degrees of artificial cold are obtained; and the

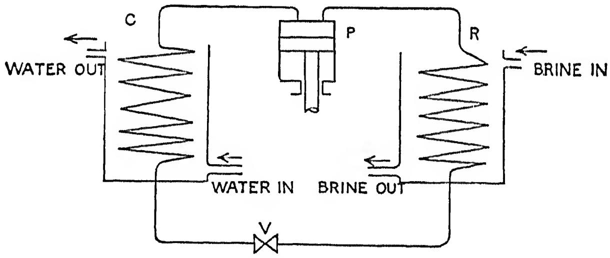


Fig. 1.—Diagrammatic Compression Refrigerating System

third makes artificial cold a *commercial* possibility by permitting the volatile liquid to be used over and over again.

The system may be illustrated diagrammatically by fig. 1, in which R is the refrigerator. Here the volatile liquid evaporates inside a coil of pipe immersed in a tank containing the substance (usually brine) to be cooled. P is the compressor which compresses the vapour produced in the refrigerator when the liquid evaporates. C is the condenser in which the compressed

gas is cooled by water and condensed into a liquid again; and V is the valve which regulates the supply of liquid from the condenser to the refrigerator. Such a system is known as a compression plant from the fact that a compressor forms an integral part of it, and it will be seen that for its operation a supply of condensing water is required and also power to drive the compressor. All compression plants must embody the essential features just outlined, although the actual details of construction vary according to the duty which the plant has to perform. It should be noted that the amount of heat to be removed in the condenser is the combined total of the amount of refrigeration done plus the heat equivalent of the power required to drive the compressor.

There is, however, a second way of attaining the desired end without using a compressor. This system is illustrated in fig. 2. In this there

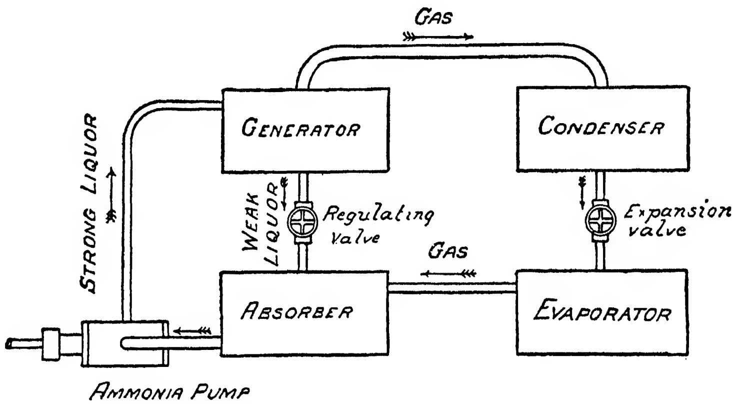


Fig. 2.—Diagrammatic Absorption Refrigerating System

is the refrigerator or evaporator in which the liquid evaporates and cools its surroundings just as in the compression system. The vapour formed is not, however, taken into a compressor, but passes into another vessel called the absorber. Here it is dissolved by a suitable absorbing liquid, forming a strong solution of the vapour in the absorbing liquid. This strong solution is then forced by the pump into a third vessel called the generator. Here the strong solution is heated and the vapour is driven off from the absorbing liquid.

Owing to this heating, the vapour is now at a much higher pressure than when in the refrigerator. From the generator the vapour passes into the condenser, where it becomes liquid, and in turn passes on to the refrigerator through the expansion valve. Such a plant is called an absorption system, due to the vapour from the refrigerator being absorbed in a liquid. It will be seen that the absorber, pump, and generator take the place of the compressor in the first system. In these diagrammatic illustrations several details essential to the proper and economical operation of a commercial plant have been left out to avoid complications. These details will be discussed later.

A number of volatile liquids for use in refrigerating plants have been proposed or tried at various times, but in present-day practice use is chiefly made of ammonia $(NH)_3$ and carbon dioxide (CO_2). Sulphur dioxide (SO_2) is also used to some extent in small machines, but it is not nearly so popular as the first two refrigerants mentioned. Ethyl chloride, a volatile hydrocarbon with the formula C_2H_2Cl, has also been used for small machines of the domestic class. These substances are all used with compression plants. For machines working on the absorption system ammonia is used, the ammonia being absorbed either in water or in ammonium nitrate. In another special design of plant water itself has been used, the cold being produced by the rapid evaporation of the water in a vacuum.

The characteristics of the various refrigerants just mentioned are generally well known, but we may say that ammonia has an extremely pungent smell, and if present in the atmosphere of a confined space is very poisonous. Ammonia also attacks copper and its alloys with great readiness, and ammonia refrigerating plants must therefore be entirely of iron. Ammonia is also expensive, and the greatest care must be exercised to prevent leaks. It is a very light gas. The liquid has a high latent heat of evaporation, a moderate specific volume, and only a moderate degree of compression is required to liquefy it in the condenser.

CO_2 has entirely different properties. This refrigerant does not attack metals, and no special arrangements are necessary in this respect. The pure gas has no smell, but in practice if it is present in the air in large quantities, as from a leak, it will be found to have a slightly acid effect on the nose, rendering it detectable.

Small quantities cannot be detected. CO_2 is very heavy, and excess of it in confined spaces will cause suffocation. It is less dangerous than ammonia. The liquid CO_2 has only a small latent heat of evaporation, which is a drawback. This, however, is counterbalanced by its small specific volume, which therefore needs a much smaller compressor than NH_3. A very high compression is required to liquefy the CO_2 in the condenser. CO_2 is much cheaper than the other refrigerants.

SO_2 when dry does not attack metals. It has a very characteristic odour, and if inhaled even in small quantities may cause serious injury. The liquid has a moderately high latent heat of evaporation, a point greatly in its favour. This is, however, offset by its high specific volume, thus requiring a large compressor. The pressure required to liquefy the SO_2 in the condenser is low. If an SO_2 machine is required to produce temperatures below 12° F., the liquid SO_2 in the refrigerator must be caused to evaporate under a vacuum. This is a disadvantage which will be dealt with more fully later on.

As regards expense, SO_2 occupies an intermediate position between NH_3 and CO_2. Ethyl chloride is not used except in small machines. As regards its physical properties it has a greater specific volume and lower vapour pressure than even SO_2. These points, however, favour its use in domestic plants, which are its field of application at present.

The following tables briefly illustrate the different physical features of the various refrigerants.

TABLE I

Properties of Saturated Ammonia *

Temp. Deg. F.	Pressure, lb. per sq. in. absolute.	Specific Vol. of Vapour, c. ft. per lb.	Heat Content of Liquid.	Latent Heat of Evaporation.	Density of Liquid, lb. per c. ft.
110	249·6	1·21	89·6	472·9	35·79
90	181·8	1·605	65·3	493·5	
70	129·2	2·296	42·1	512·8	37·85
50	89·09	3·278	19·8	531·0	
30	59·35	4·82	– 1·9	548·1	
10	38·02	7·34	– 23·2	564·4	40·61
– 10	23·3	11·63	– 44·2	579·9	
– 30	13·56	19·35	– 65·0	594·7	42·3

TABLE II

Properties of Saturated Carbon Dioxide *

Temp. Deg. F.	Pressure, lb. per sq. in. absolute.	Specific Vol. of Vapour, c. ft. per lb.	Heat Content of Liquid.	Latent Heat of Evaporation.	Density of Liquid, lb. per c. ft.
88·43	1071·0	0·0346	59·23	0	28·9
60	744·2	0·0992	16·93	76·14	
40	565·4	0·1446	4·367	94·13	
20	422·0	0·2058	– 6·102	107·3	58·95
0	308·6	0·2918	– 15·41	117·7	
– 20	220·6	0·4166	– 23·96	126·0	64·34

TABLE III

Properties of Saturated Sulphur Dioxide *

Temp. Deg. F.	Pressure, lb. per sq. in. absolute.	Specific Vol. of Vapour, c. ft. per lb.	Heat Content of Liquid.	Latent Heat of Evaporation.	Density of Liquid, lb. per c. ft.
100	84·4	0·96	23·1	140·9	100
80	59·3	1·37	16·1	146·4	
60	40·4	1·96	9·3	151·1	
40	26·7	2·88	2·6	155·4	100
20	17·0	4·42	– 3·8	159·1	
0	10·4	7·12	– 9·9	162·0	
– 20	5·9	12·02	– 15·9	164·9	100

* See works referred to on p. 202 for full and complete tables of properties.

TABLE IV

PROPERTIES OF SATURATED ETHYL CHLORIDE *

Temp. Deg. F.	Pressure, lb. per sq. in. absolute.	Specific Vol. of Vapour, c. ft. per lb.	Heat Content of Liquid.	Latent Heat of Evaporation.	Density of Liquid, lb. per c. ft.
100	34·8	2·6	29·2	159·8	54·2
80	24·5	3·6	20·7	163·7	55·2
60	16·6	5·2	12·0	167·5	56·25
40	10·6	7·8	3·4	171·5	57·25
20	6·7	12·0	— 5·4	174·7	58·3
0	4·0	19·0	— 13·9	177·8	59·3
— 20	2·4	32·5	— 22·3	180·6	60·3

A point of importance must now be discussed, namely, the action which takes place as the refrigerating liquid is transferred from the condenser to the refrigerator through the valve V, fig. 1.

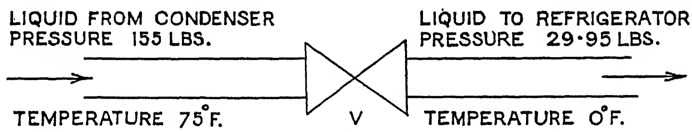


Fig. 3

To make our ideas definite we will consider an ammonia system whose condenser is supplied with cooling water at a temperature of 70° F., and whose refrigerator is working at a temperature of 0° F., the pressure in the refrigerator being therefore 29·95 lb. per square inch absolute.

Fig. 3 represents a small portion of fig. 1 in the neighbourhood of the valve V to a larger scale. The pipe on the left-hand side brings the liquid from the condenser, while the pipe on the right-hand side takes it to the refrigerator. Now as the condenser is supplied with cooling water at 70° F. the pressure in the condenser would be about 155 lb., and the ammonia gas from the compressor would be condensed to a liquid and cooled to say 75° F. Under such conditions (pressure 155 lb. and temperature 75° F.) it arrives at V. Passing this valve it arrives on the refrigerator side, where the pressure is 29·95 lb., and under this pressure the liquid ammonia boils away at a temperature of 0° F. Now the mere fact of the liquid passing through the valve V does not alter its temperature, and it therefore arrives at the refrigerator side of V with a temperature of 75° F. (under our assumed conditions), that is to say, much above its boiling-point. The liquid therefore commences to boil away, abstracting heat in doing so. It is quite clear, however, that before it can do any useful refrigeration and remove heat from its neighbouring objects, it must first cool itself down to the refrigerator temperature (in our case 0° F.). This is just what

* See works referred to on p. 202 for full and complete tables of properties.

happens. When we allow the liquid refrigerant to pass through the valve v from the condenser to the refrigerator, it immediately begins to evaporate, that portion so evaporating cooling the remaining part of the liquid down to the refrigerator temperature, when this part in turn begins to do *useful* work. It will be quite clear that the hotter the liquid leaving the condenser the more of it we must use to cool itself, and therefore the less remaining for useful refrigeration.

The amount so used up also depends on the particular refrigerant we are using, and therefore we should work with a refrigerant which does not use up much liquid for this purpose, and cool that liquid as much as possible before sending it into the refrigerator. The following figures show the amount of liquid used up to cool itself down to the refrigerator temperature with the various refrigerants.

TABLE V

LOSS OF REFRIGERATING EFFECT DUE TO HEAT IN THE REFRIGERANT LEAVING THE CONDENSER

Temperature of Refrigerant leaving Condenser.	Loss of Available Refrigerating Effect.		
	SO_2.	NH_3.	CO_2.
95° F.	19 per cent	19 per cent	64 per cent
85 ,,	17 ,,	17 ,,	52 ,,
75 ,,	15 ,,	15 ,,	38 ,,
65 ,,	13 ,,	12 ,,	31 ,,
55 ,,	11 ,,	10 ,,	25 ,,

Refrigerator temperature 0° F. in all cases.

It will be observed that the loss with CO_2 is very heavy under certain conditions, and means of overcoming this will be described in the section on machinery.*

CHAPTER I

Machinery

Considering the machinery itself it may be divided into two classes, namely, land and marine. There is a general similarity between machines of different makers, although differences arise in the smaller details of construction. Most compressors are of the reciprocating type, but rotary machines are now receiving attention.

* For complete tables of properties of the various refrigerants see: Ammonia, *The Properties of Saturated and Superheated Ammonia*, by Goodenough and Mosher; CO_2, "The Properties of Saturated CO_2", by G. C. Hodsdon, in *Ice and Cold Storage*, November, 1912; SO_2, *The Elements of Refrigeration*, by A. M. Greene; Ethyl Chloride, "The Properties of Ethyl Chloride", by G. C. Hodsdon, in *Ice and Cold Storage*, December, 1919.

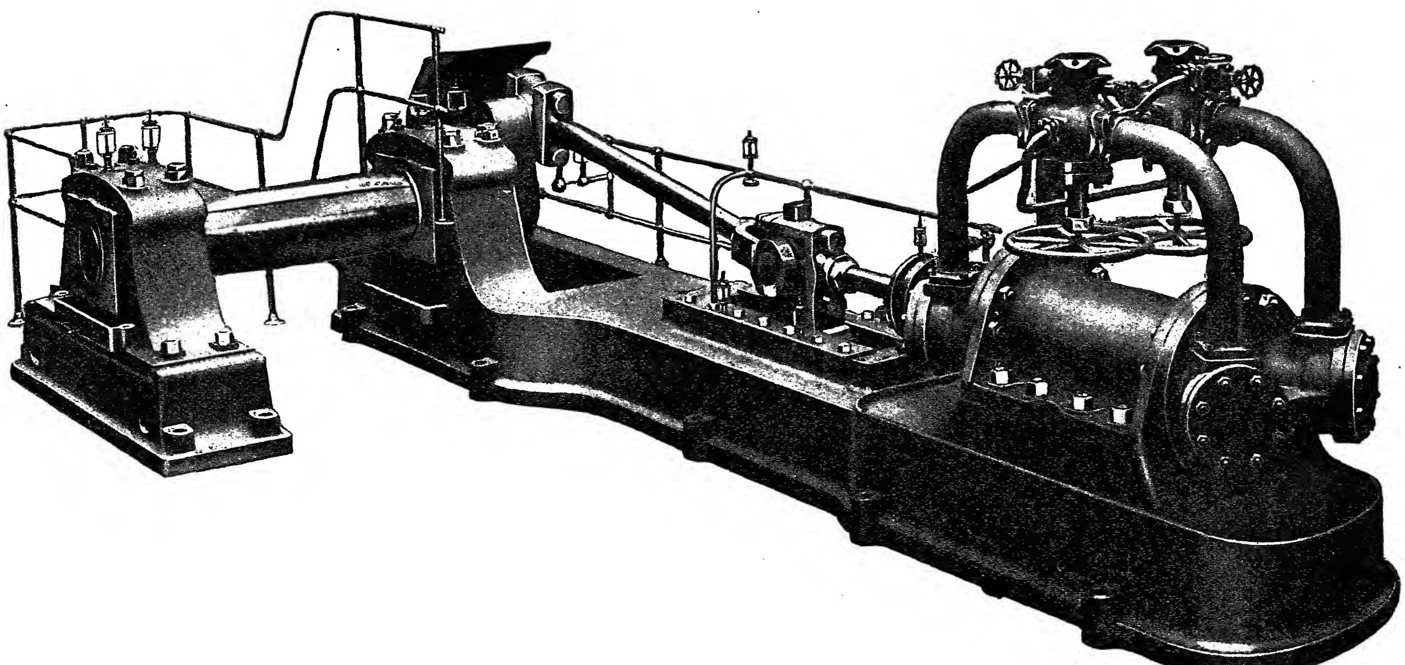


Fig. 4.—Horizontal Land-type Ammonia Compressor

Up till recently the practice in this country has been to use a single-stage compressor with large poppet valves. These valves necessitate a low piston speed for their satisfactory operation, and in consequence the compressors are of large size. The heat developed during compression is taken care of by carrying over some of the cold liquid refrigerant from the evaporator, and the compressor is not water-jacketed. American practice, on the other hand, is to bring dry gas back to the compressor and rely on a water jacket to keep the machine from overheating.

The piston speed is about 200 to 350 ft. per minute, but recent developments in the shape of light steel-plate valves have put the piston speed up to 500 to 550 ft. per minute. This has resulted in a much smaller machine for a given duty.

The majority of machines are on the ammonia system, and fig. 4 shows a typical land machine arranged for belt driving. The connection between the piston-rod and crosshead should in all cases be adjustable, as it is usual to run with a very small linear clearance ($\frac{1}{32}$ in.) between the piston and cylinder cover, to secure the highest efficiency. Fig. 5 shows a section of a compressor cylinder. The piston is packed with plain cast-iron rings. It is in the design of the valves that most difference occurs between different makes. These are usually made of a tough steel with cast-iron seats, and work in a long guide. A common fault with most valves is that they are given too much lift and too great an area of contact with their seats in the first instance, and the author has found that both may be reduced with good results. The suction valve opens into the cylinder, and if the stem breaks, the valve head is liable to fall into the cylinder and cause a breakdown. Fig. 5 shows a typical suction valve in which this is obviated by a nut screwed and pinned on the spindle. The spindle end is reduced in diameter and screwed to take a nut which regulates the lift and takes the shock caused by the beating of the valve. Alternatively a collar solid with the spindle may be used. With this design the valve seat and guide must be split longitudinally to get the valve into place, and this joint must be carefully machined to avoid leakage. The figure shows also a typical delivery valve. In normal working the lift of the valve is entirely controlled by the spring A. Should, however, an excessive amount of liquid ammonia enter the cylinder, the heavier buffer spring B permits of additional lift being obtained automatically, and the excess liquid is forced out of the compressor without damage. Compressor rods are made of a hard steel (tensile strength 40 tons per square inch) and should be ground true. The very long stuffing box used is shown also. The lantern dividing the box into two parts will be noticed, and this lantern space is connected to the compressor suction to reduce the pressure against which the packing has to hold tight. If the compressor is driven by anything other than a steam-engine, a bypass connection between the suction and delivery should be fitted to ensure easy starting. Pumping-out connections should also be fitted. A dirt trap should be fitted on the suction side of the compressor to intercept scale and dirt, and an efficient oil separator on the delivery

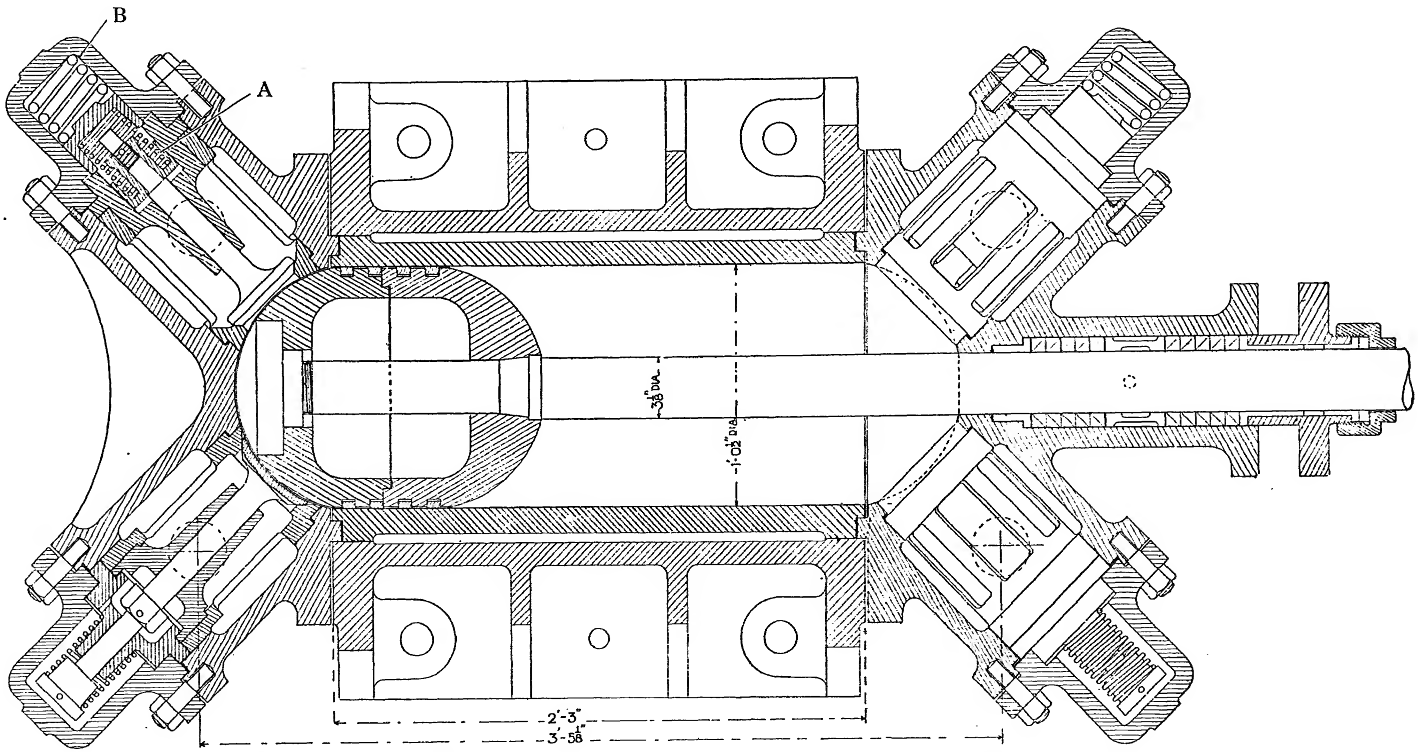


Fig. 5.—Arrangement of Compressor Cylinder Piston and Valves for $12\frac{1}{2}$ in. × 21 in. Triplex Ammonia Compressor

side. Two kinds of separator are in common use. In one the gas is given a whirling motion to remove the oil by centrifugal force. In the other a sudden reduction in velocity is relied on to throw the oil out of suspension. In some cases a water-cooled separator is used.

Some ammonia plants have a liquid receiver. This is simply a strong vessel in which the liquid ammonia from the condenser is stored before passing to the refrigerator. If any oil is carried over from the compressor it has a tendency to settle in the receiver, and this vessel is provided with an oil drain valve.

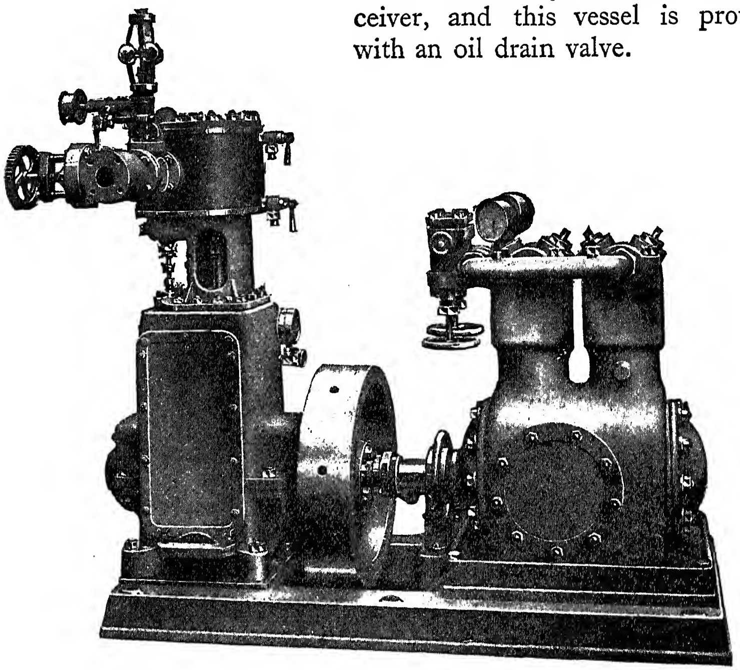


Fig. 6.—High-speed Vertical Single-acting Ammonia Compressor

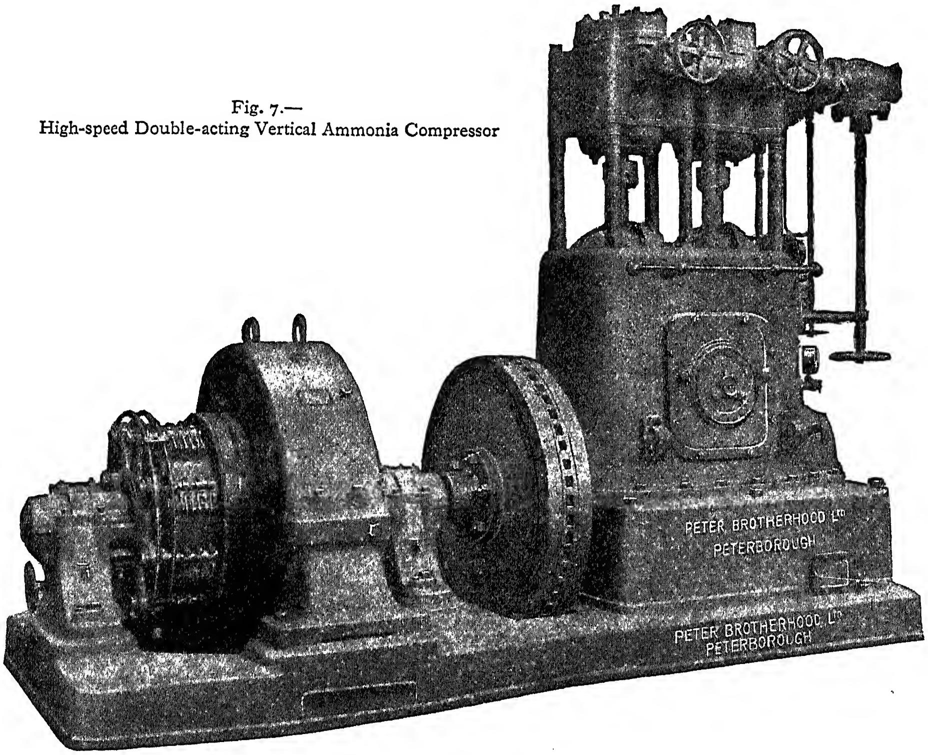
Smaller-sized machines are now made vertical with an enclosed crank-case, all moving parts running in oil. The pistons are single acting, of the trunk type, thus dispensing with glands and compressor rods. The valves are in the cylinder cover, of similar design to those previously described. Such machines are made with one, two, three, or four cylinders. They run at higher speeds than the horizontal type, and are becoming universal for small plants. The crank-cases must be strongly designed, as in the event of leaky valves or piston rings considerable pressure may come upon them. A relief pipe must also be fitted to draw off the ammonia before opening the case for cleaning. The latest development in this type of compressor has shoes on the piston to take the thrust of the connecting rod, and rings to prevent oil creeping up the cylinder. The valves are in the form of very light steel alloy plates. Very good results have been obtained with them. These machines run at much higher piston speeds

than previously attempted, and mark one of the most radical advances of recent times. Fig. 6 shows a single-acting machine of this kind, and fig. 7 a double-acting machine. Fig. 8 shows a high-speed horizontal compressor of excellent design. The distinctive construction of the cylinder will be noticed, and also the very complete system of forced lubrication.

The discharge temperature is usually about 120° to 140° F., but this may be carried up to 180° F. with care without detriment.

With regard to marine machines the most notable feature is the very complete precautions taken to prevent total breakdown. Fig. 9 shows a

Fig. 7.—
High-speed Double-acting Vertical Ammonia Compressor



large machine for carrying meat cargoes, and has three compressors driven from the tail rods of a triple-expansion engine. The crank-shaft is in three interchangeable parts, and the steam connections are arranged so that any two cylinders can work together or any cylinder can work by itself.

Fig. 10 is a small steam-driven machine with one compressor, and has its condenser coil in a cast-iron casing at the back of the machine. A condenser water pump is driven off the end of the crank-shaft. This type of machine is used for ships' provision chambers.

The principle of action of CO_2 machines is precisely the same as ammonia machines, but the details of construction are very different, owing to the different physical properties of CO_2. A CO_2 compressor cylinder is usually made from a steel block to withstand the very heavy pressures to which it

is subjected. The bore of the cylinder and the size of the valves are less than in the case of the ammonia machine of equal duty. Also the valves do not project into the cylinder, but are in pockets at the side. The CO_2 machine is thus safer in this respect than the ammonia machine. The

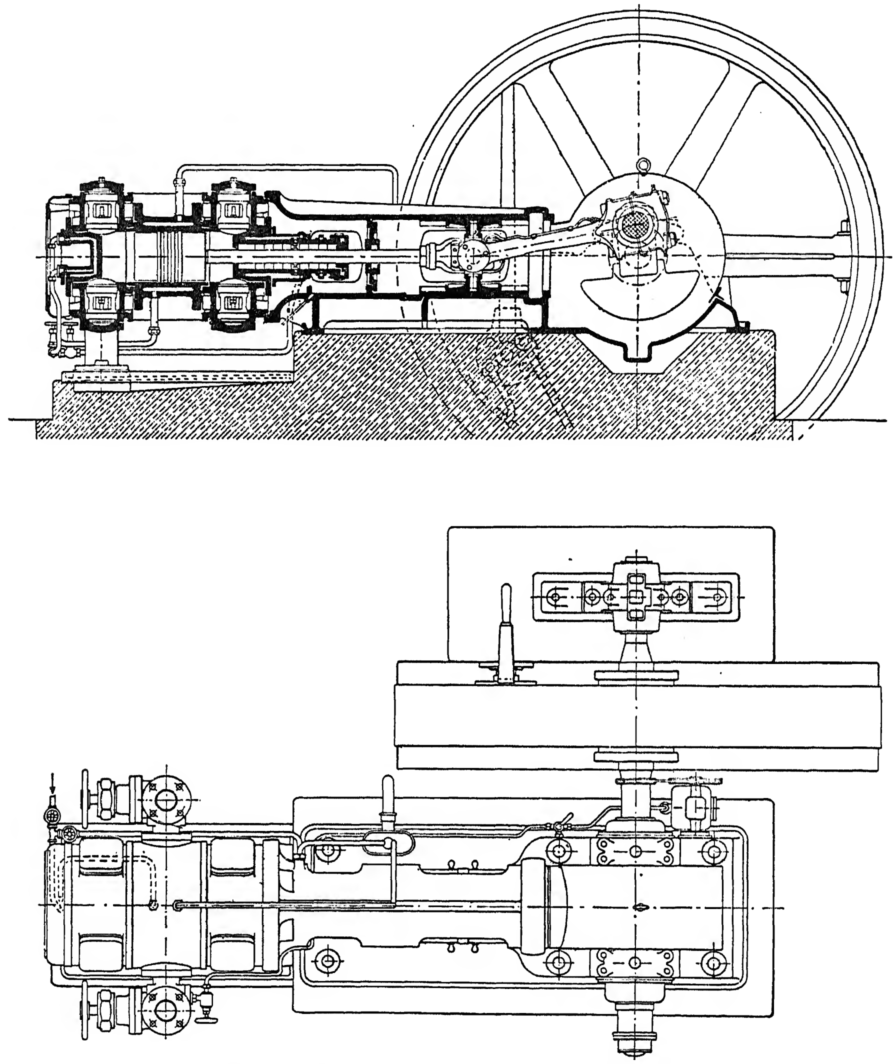


Fig. 8.—High-speed Horizontal Double-acting Compressor

piston is commonly packed with hat leathers, but greater use is now being made of cast-iron rings. This is a great improvement, as the rings do not deteriorate as the leathers do if the compressor becomes too warm. The gland is packed with U leathers (usually made of Woodite or similar material) fitted on either side of a lantern, and oil pumped into the space between the leathers. The arrangement is often more effective on paper than in service,

and it is frequently hard to keep the oil pumps in use without excessive consumption of oil. Consequently many attempts have been made to design a wholly metallic packing, and with considerable success. Fig. 11 is a large horizontal machine, double acting and driven through gearing. The ends of the cylinder are in this case connected by steel blocks, but a more extended use of welded fittings is now being made.

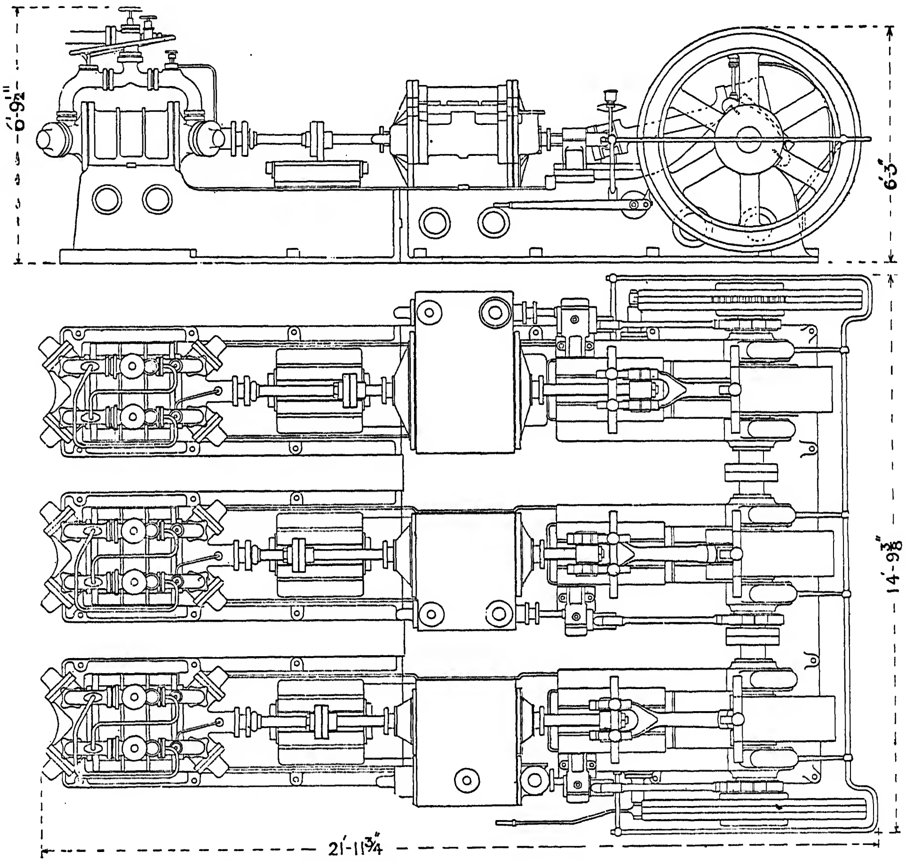


Fig. 9.—General Arrangement of 12½ in. × 21 in. Triplex Ammonia Compressor

Fig. 12 shows a duplex marine machine with the condenser coils in the box-bed.

With regard to SO_2 machines, these are only used in small sizes, and the design follows closely that of ammonia machines. The pressures in a SO_2 system are lower than in the ammonia or CO_2 systems. Indeed if a temperature of 12° F. is being carried, the compressor suction is less than atmospheric, and consequently there is a risk of air being drawn into the plant through the gland. The moisture which this air takes in may freeze up the plant. SO_2 is a good lubricant and no oil is required, but if SO_2 becomes wet it may set up corrosion.

The absorption system has been illustrated diagrammatically in fig. 2. The generator consists of a steel shell with strong steel flanges securing end covers. The shell is fitted with a system of internal tubes opening into a header on the end cover. A strong solution of ammonia in water is placed in the generator, and live steam (or exhaust) passed through the tubes. This drives off the ammonia in the form of gas with some water vapour.

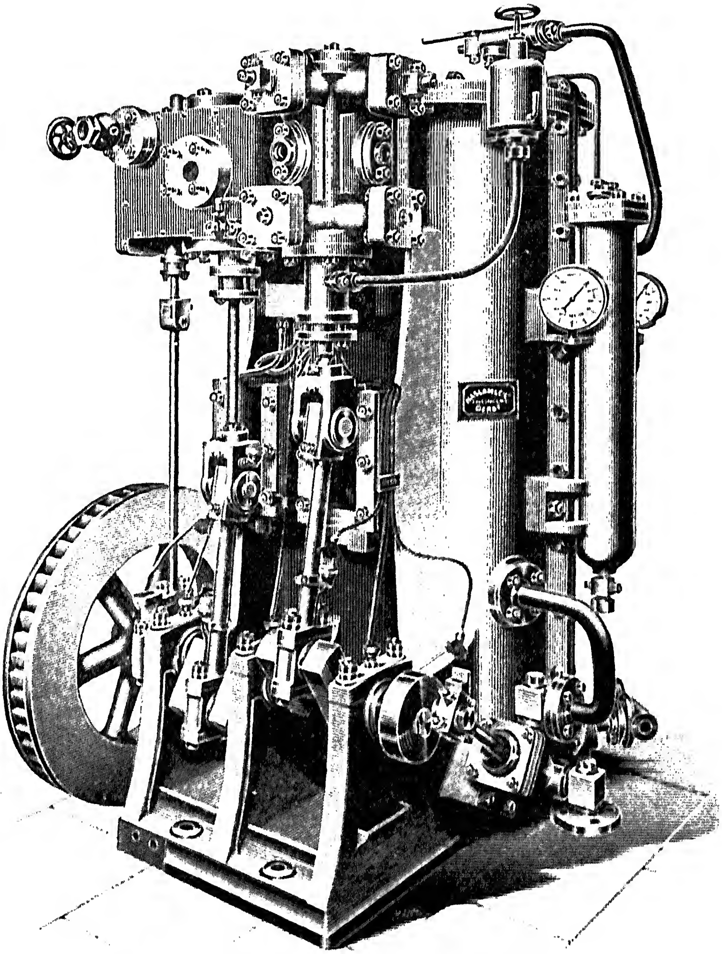


Fig. 10.—Small Marine-type Ammonia Compressor

This gas passes to the analyser, where it comes in contact with the much cooler liquor being fed into the generator, resulting in most of the entrained water vapour being condensed and carried back to the generator. The gas goes onwards to the rectifier in a much dryer condition. The rectifier may consist of coils cooled by water. Here the remainder of the moisture is condensed out, and the ammonia goes to the condenser in practically an anhydrous form. The ammonia is condensed to a liquid, run into a receiver, and fed to the refrigerator as required. The gas formed in the refrigerator

is led away to the absorber, a vessel whose construction is very similar to the generator. The liquor in the generator, after having some of its ammonia

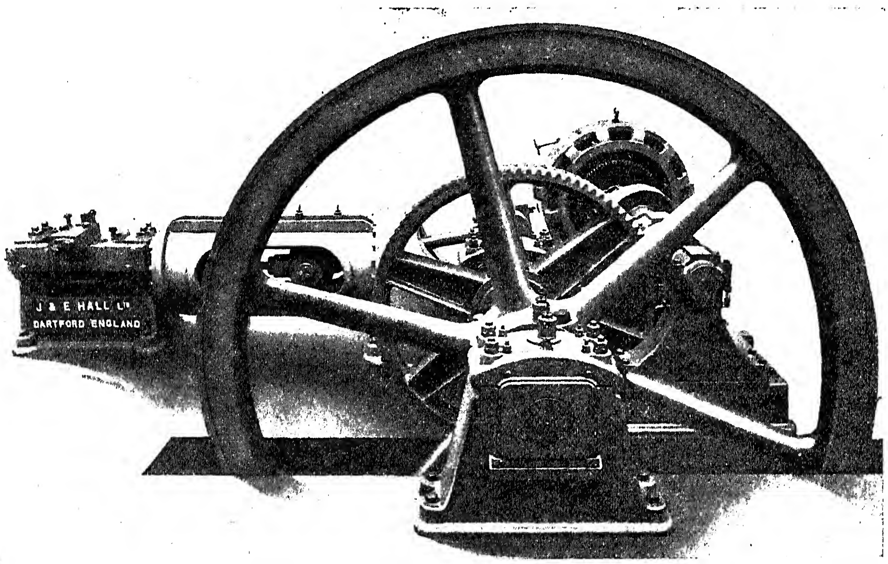


Fig. 11.—Large Horizontal Land-type CO_2 Compressor

driven off, runs from the generator to the absorber, passing on its way through the interchanger. In the absorber this liquor absorbs the gas from the

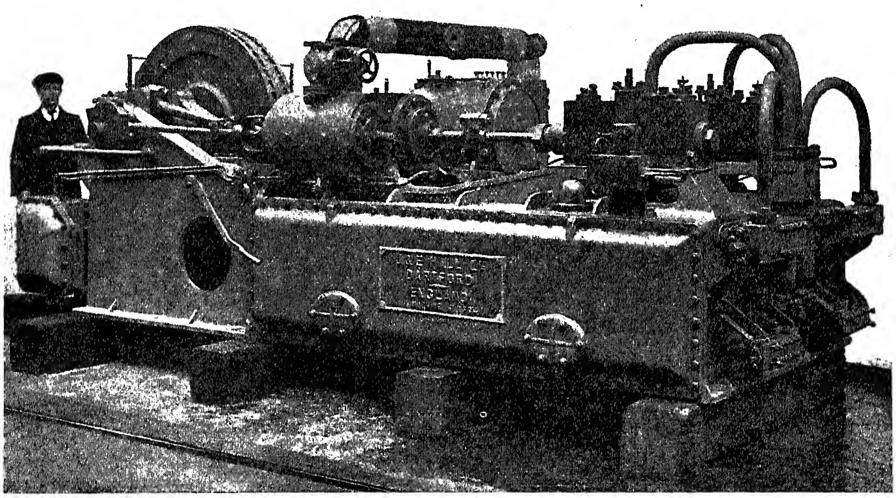


Fig. 12.—Duplex Marine-type CO_2 Compressor

refrigerator and is then returned to the generator for further heating. As heat is generated during the absorption it is found necessary to cool the

absorber by circulating water through the tubes. The pressure in the absorber is about 5 to 10 lb. less than in the refrigerator. The liquor is pumped from the absorber to the generator. The pump is specially designed, no brass or copper being permissible in its construction. The interchanger previously mentioned is fitted between the absorber and generator to economize heat during the operation of the plant. The construction is usually a series of concentric tubes, the comparatively cold liquor from the absorber flowing through one whilst the hot liquor from

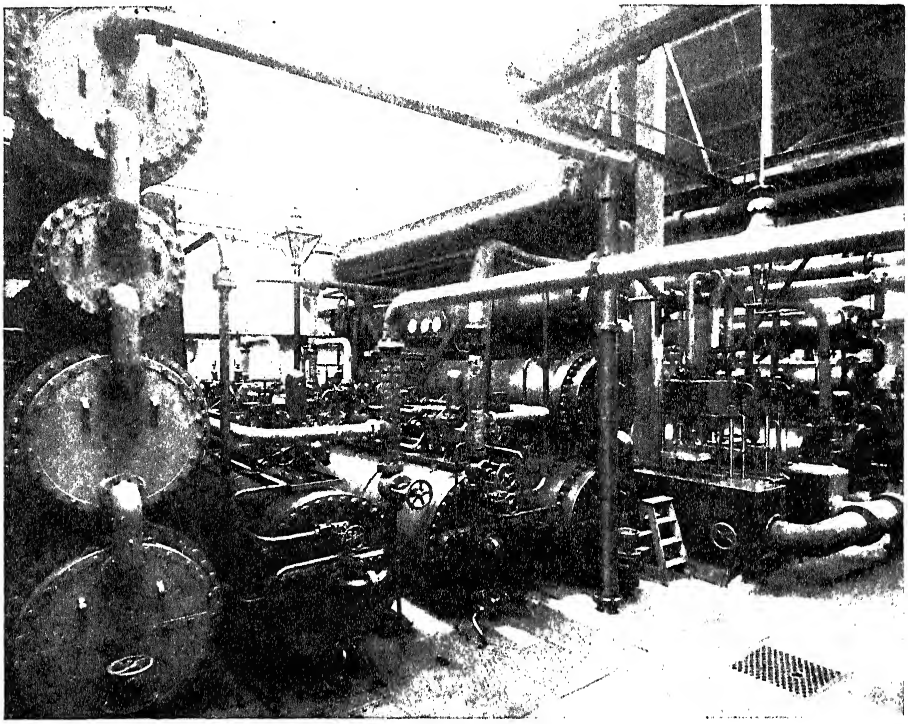


Fig. 13.—Absorption Refrigerating Plant

the generator flows through the other. An interchange of heat takes place, the strong liquor arriving at the generator hotter than it would otherwise do, and requiring less steam to heat it. The weak liquor reaches the absorber cooler than it would otherwise do, and thus requires less cooling during absorption.

An outstanding feature of the absorption system is its ability to work with waste heat in the form of exhaust steam. In fact there are plants at work which only require the exhaust steam furnished by the liquor and brine pumps. They require, however, more cooling water than a compression plant, and must be designed by experienced persons, or there is a liability of moisture being carried over into the low-temperature side of the plant and freezing it up. The absorption system has a disadvantage

ring time to adjust itself to changes in operating conditions owing
irge masses of liquor which have to be heated and cooled. It is
: not so flexible as the compression system. Undoubtedly, how-
ere waste steam is available it is the most economical, and its value
e more appreciated than hitherto. Fig. 13 is a large absorption
general arrangement, entirely worked by the exhaust steam from
or and brine pumps and electric-lighting engines.
ier absorption system has lately been evolved, but has not taken
in commercial work. This is the Seay process, in which the
is absorbed not in water but in ammonium nitrate. Very economi-
s are claimed for this plant, but as the vessels have to be constructed
ied with alu-
ts first cost is

oss occasioned
cessity of the
frigerant hav-
to cool itself
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liagrammatic-
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o overcome
is far as pos-
he system is
the Multiple
Compression.

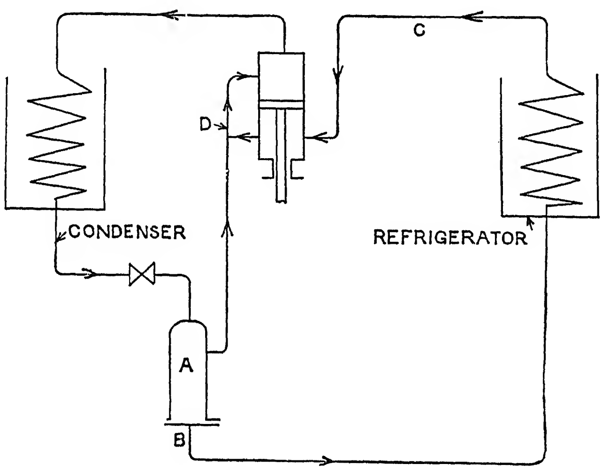

Fig. 14.—Multiple Effect Compression System

d leaving the condenser passes through a regulating valve into
A, in which the pressure is maintained at some intermediate
ween that in the condenser and that in the refrigerator. A
nount of expansion takes place in passing through this valve
receiver, and a portion of the liquid evaporates, cooling down
nder to the temperature corresponding to the pressure in the
The receiver A has an outlet for the cooled liquid controlled
-operated regulator valve B, which allows the liquid to pass into
erator. The expansion past the valve B causes more of the
evaporate and cool down the rest to the refrigerator temperature.
quid approaching the valve B has already been partially cooled
re is less evaporation at expansion into the refrigerator than is
e case. Consequently the refrigerator surface is in contact with
proportion of liquid than usual, and thus does more and better
ie gas formed in the refrigerator is taken through the pipe C into
f the compressor and compressed to the pressure in the receiver.
drawn into the other end of the compressor through the pipe D,

together with the gas from the receiver A. The whole is compressed to the full pressure and discharged into the condenser. The operation resolves itself into expansion in two stages and compression of the gas in two stages. The improved action is obtained by each pound of the liquid fed into the refrigerator having a greater refrigerating effect due to its preliminary cooling. Also owing to this cooling being done by the liquid itself at a somewhat higher pressure than that in the refrigerator, the gas formed in the receiver can be taken into the compressor without the latter having a greater swept volume than necessary for the gas from the refrigerator alone. More total power is required to operate the machine, but the power per unit of refrigeration obtained is less. These machines are also being made double-acting, and in some cases the float-operated valve is dispensed with and worked by hand.

Many other systems have been devised to accomplish the same end as the Multiple Effect, but owing to lack of space they cannot be dealt with here. Further information can be obtained from the *Proceedings* of the Cold Storage and Ice Association; also the Liverpool Engineering Society, vol. 40.

Tables VII and VI give the amounts of ammonia and CO_2 to be circulated to produce 1000 B.Th.U. of refrigeration per 24 hours under various conditions. These tables take account of the loss of refrigerating effect by expansion at the regulating valve, and also the loss due to using the liquid refrigerant to keep the machine cool. They do not take account of the loss due to the clearance and throttling in the compressor.

For hot climates or for very low refrigerator temperatures compound compressors are used in which the gas is compressed in two stages. These are quite distinct from those multiple-effect machines which adopt two-stage compression.

Condensers are in the main of three general types:

1. Submerged;
2. Atmospheric or surface evaporative;
3. Double pipe.

The submerged type consists of circular or oval coils of pipe placed in a tank through which water circulates. Condensers of this type are usually of 1 in. or $1\frac{1}{4}$ in. nominal bore tube. The lengths of pipe composing the coil are welded together so that each coil is jointless. The various coils are nested, that is, placed one inside the other, and secured to flat bar stays by U clips, so that the whole is made rigid and the coils properly spaced apart. The coil ends are connected to headers or boxes. These are of cast iron or built up of welded pipes for ammonia, while for carbon dioxide welded pipe headers or mild steel blocks are used. For CO_2 condensers solid-drawn steel pipe is preferable, or frequently copper pipe about $\frac{7}{8}$ in. bore is used, as this is not attacked by sea-water and the coils have a good scrap value when finished with. Iron coils should always be galvanized outside when used with sea-water. All coils should be tested by hydraulic pressure to ensure safety, and by compressed air when submerged under

water to guard against porosity and pin-holes. The submerged condenser demands copious water-supply, the water going to waste after passing through the tank. In most situations on land water must be economized, and this led to the introduction of the atmospheric or surface evaporative condenser. This consists of a number of grids of pipes, interlaced to give close spacing, and secured to vertical standards over a tray or shallow tank. A slotted pipe runs over the top of each sheet of condenser pipes and is supplied with water from a central water box. The water trickles downwards over the coils, and is collected in the tray and circulated over again. These condensers are placed in the open air, so that some of the water evaporates, thus cooling the remainder and enabling it to be used over again. For the same duty this type requires more surface and a much larger quantity of water to be in circulation. The ammonia gas enters at the top and the liquid leaves at the bottom. Both the ammonia and water flow in the same direction (downwards), which does not give the greatest rate of heat transmission through the tubes. Further, the liquid ammonia about to leave the condenser is in contact with the hottest water about to fall into the tray, and is therefore not cooled to the greatest extent.

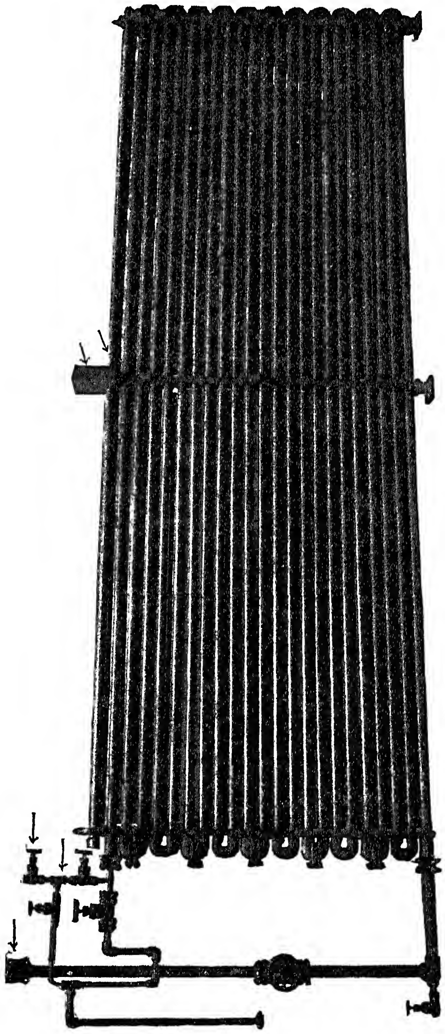

Fig. 15.—"Flooded" Ammonia Condenser

To obviate these drawbacks the condenser shown in fig. 15 was introduced and is called a flooded condenser. The ammonia gas enters at the bottom whilst the liquid leaves at the top, the ammonia and water flowing in opposite directions. Various patented forms of this condenser have been put forward which possess no advantages over the simple type. Flooded condensers must be carefully designed or trouble will arise.

In some cases, particularly in small installations, it is desirable to have a condenser with a closed water system, in which the water is always con-

tained in pipes. In such cases the double-pipe type is used. This consists of a series of concentric tubes connected at the ends by special castings and bends. The cold water circulates upwards through the inner tube, whilst the ammonia passes downwards through the annular space between the tubes. The heat transmission is good, but there are a large number of joints to keep tight.

Refrigerators are generally very similar in type to condensers.

Brine coolers of the double-pipe type have been used in some cases.

For brine cooling on board ship the refrigerator is practically a replica of the submerged condenser, the surface being suitably proportioned to meet the altered conditions of working. Coils in contact with brine should not be galvanized.

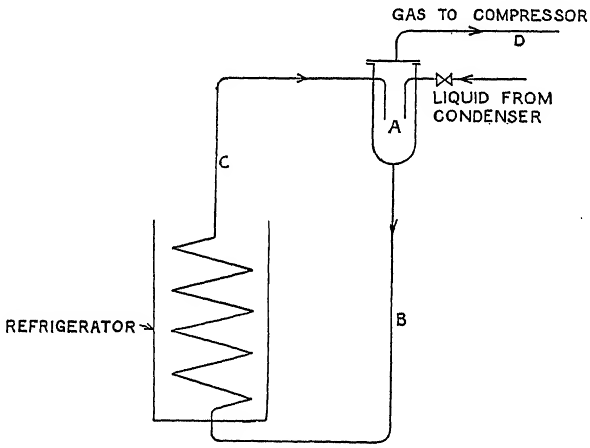

Fig. 16.—" Flooded " Refrigerator System

For land work the refrigerator is arranged to cool air in many cases. In such instances it is built of vertical sheets of pipes very similar to the atmospheric condenser but spaced more closely together. The whole is arranged in an insulated casing and air circulated over the coils by a fan. Moisture is deposited out of the air. The insulation must therefore be waterproof. Brine is frequently circulated over the coils to prevent undue accumulation of frozen moisture on the pipes. This brine requires regular boiling to keep up its strength or it will itself freeze on the coils.

With the refrigerators just described the usual practice is to introduce the liquid refrigerant at the bottom coil and withdraw the gas at the top. A flooded refrigerator which gives greater efficiency has been introduced, and is shown diagrammatically in fig. 16. A separator vessel A is placed at a suitable height above the cooling coils and the liquid refrigerant fed into it. A pipe B connects the separator to the bottom of the coils, and another pipe C connects the top of the coils to the separator. Pipe D connects the separator to the compressor. Any gas formed during expansion through the regulating valve immediately returns to the compressor, and liquid only enters the coils through pipe B. The mixture of gas and liquid leaving the coils again separates in the vessel A, the gas going to the compressor and any liquid returning to the coils. The fullest possible use is thus made of all the liquid. Care must be taken in introducing such a system in English practice, as with a flooded refrigerator dry gas returns to the compressor. Means must therefore be provided to prevent overheating of the compressor,

either by fitting a water jacket or by introducing a small jet of liquid refrigerant into the compressor suction. The author's experience with the latter method indicates that it is very superior to the water jacket, and it can further be applied to existing machines without trouble.

The following figures indicate the usual allowances for pipe surfaces in condensers and refrigerators:

Submerged Condensers.—4¾ to 5½ lineal feet of 1¼-in. pipe per 1000 B.Th.U. of refrigeration per hour, or 9¼ lineal feet of 1-in. pipe.

Atmospheric Condensers.—9½ lineal feet of 1¼-in. pipe per 1000 B.Th.U. of refrigeration per hour, or 11 to 11¼ lineal feet of 1-in. pipe.

Brine Coolers.—10 to 12 lineal feet of 1¼-in. pipe per 1000 B.Th.U. of refrigeration per hour, or 14 to 14½ lineal feet of 1-in. pipe.

Flooded Condensers require only about one-third the surface of ordinary atmospheric condensers.

The above are practical rules and they should be used with discretion, particularly where the condenser pressure is high or the refrigerator pressure is low.

TABLE VI

CUBIC FEET OF CO_2 GAS TO BE CIRCULATED PER HOUR TO PRODUCE 1000 B.TH.U. OF REFRIGERATION

		Condenser Conditions.			
Pressure lb. per sq. in. ..		—	—	1100 lb.	1250 lb.
Water temperature		60° F.	70° F.	80° F.	90° F.
Refrigerator Temperature.	30° F.	2·3	2·7	4·3	4·9
	20° F.	2·7	3·2	4·9	5·6
	10° F.	3·1	3·8	5·9	7·1
	0° F.	3·7	4·5	6·9	9·0
	− 10° F.	4·4	5·3	8·7	11·7

TABLE VII

CUBIC FEET OF AMMONIA GAS TO BE CIRCULATED PER HOUR TO PRODUCE 1000 B.TH.U. OF REFRIGERATION

		Condensing Water Temperature.			
		60° F.	70° F.	80° F.	90° F.
Refrigerator Temperature.	30° F.	9·6	9·8	10·0	10·5
	20° F.	11·8	12·1	12·5	13·2
	10° F.	14·7	15·2	15·9	16·8
	0° F.	18·6	19·9	20·4	21·3
	− 10° F.	24·2	25·4	26·4	28·1

CHAPTER II

Insulation

All cold rooms, tanks, &c., in which the low temperature is produced or used must be adequately insulated if the cost of maintaining the necessary degree of cold is not to be prohibitive.

It is in practice impossible to get an ideal and perfect insulator, and a compromise has to be effected. In considering the value of an insulating material many points must be kept in mind as well as its insulating efficiency. It must not be too costly; it should be of ready application and possess considerable mechanical strength to withstand rough usage; it should be water- and fire-proof and impervious to vermin, and for marine use it should not be too heavy. It will readily be realized that it is not easy to find a substance which complies with all the foregoing requirements.

Insulation generally can be divided into three kinds:

1. Loose filling retained by boards;
2. Slab insulation;
3. Plastic insulation.

The first system is the earliest and at present the most extensively used. With this a number of grounds are secured to the surface to be insulated. A lining of T. and G. boards is then nailed to the edges of these grounds horizontally, leaving a space between the surface and the boards which is filled up with the insulating medium. Over the outer surface of the boards is tacked a layer of waterproof paper, and then a second layer of boards is put on. All surfaces are treated in a similar manner. All girders, columns, or other projections into a cold chamber must also be insulated; but if the full thickness of insulation was carried round these in all cases the storage space would be unduly encroached upon. We therefore frequently find such obstacles covered by a thinner insulation of hair felt protected on the outside by wood casing. The face of the woodwork should be painted, enamelled, or shellac varnished. The usual insulating materials used with this system are silicate cotton and granulated cork. Numerous test figures are advanced by the manufacturers of the various materials, each claiming the advantage, but these figures must be accepted with caution. In practice it is extremely difficult to keep any insulation composed of loose filling dry, and once the insulation becomes wet it is quite useless as an insulator. The importance of this point is not yet fully realized, and the author is frequently coming across cases of insulation having become useless through absorption of water. Granulated cork is less absorbent than silicate cotton. A further disadvantage of any loose filling is its liability to settle down with vibration, thus leaving empty spaces or voids through which heat readily enters the cold room. No loose fillings are vermin-proof, and in time the wood linings become wet and absorb odours which are transmitted from

one class of goods to another. Altogether such systems cannot be said to meet present-day requirements and sanitary ideas.

An improvement on this system is the use of cork slabs. These are made by warming and compressing granulated cork so that the granules adhere together, giving a slab of some mechanical strength. Such slabs are made in thicknesses of 2, 3, and 4 in., and are built up in layers to give an insulation of the desired thickness. The first layer is stuck on to the surface to be insulated by bitumastic or by a screed of Portland cement. Subsequent layers may be attached by the same means or may be nailed to the first layer. The surface of the last layer is, on the walls and ceiling, faced with cement worked to a smooth surface with the idea of giving a waterproof and sanitary finish. If, however, this coating is only of Portland cement this ideal is not attained, and in the best practice a special facing is put over the Portland. The floors are finished either in granolithic or asphalt, and the floor covering is carried 6 in. up the walls to form a skirting. Such a room may be readily washed down without moisture affecting the insulation, and to facilitate this the floors should be laid with a fall. A modification of this system is now on the market in which the last layer of slabs already has a facing attached thereto. This reduces the labour of facing in situ, but leaves a joint between each slab which is objectionable.

Such insulation is readily applicable to land work, but is not so suited for marine work. A ship is a very irregularly shaped structure with many awkward projections on the inside. It is clearly impossible to fit cork (or any other previously prepared) slabs exactly to such a structure, and any attempt to do so would result in excessive cost, as all cutting must be done by skilled labour. The result is that cavities are left in the insulation to which water may obtain access from leaking seams and rivets, and set up corrosion of the ship's structure. Some makes of cork slab also disintegrate, and recent cases have occurred with holds insulated with cork slabs and asphalt where the insulation has compressed and sunk. On investigation these have mostly been found to be due to defective cork slabs. Some of the specially faced slabs previously mentioned have also been found to buckle after erection, a slab less than 2 ft. 6 in. square warping by 2 in. This forms cavities in the insulation, and is particularly objectionable in overhead work where rails for hanging meat may be fitted.

Plastic insulation is a very promising form for future development. The principle of such is the mixture of granulated cork with a binding material, the whole being applied to the surface to be insulated in a plastic condition and allowed to set. The face of the insulation is then finished with a special facing as previously described for cork slabs. Fig. 17 shows a part section of a ship insulated on this system. Grounds are attached to the frames at intervals. The insulation is applied and allowed to take on an initial set. A light expanded metal is then tacked on to the face of the grounds and the facing applied to this. This results in a waterproof and sanitary job, and the reinforcement gives strength and flexibility to

the facing, so that it is free from troubles due to cracking. The binding material used has a distinctly preservative action on the ship's structure. Wood grounds may be dispensed with, and the reinforcement secured to rods or tubes running through the frames and beams.

A further recent development in insulating material is the Moler brick and slab made by the Moler Fireproof Brick Company. These slabs are made from a mixture of diatome silica and clay, pulped and baked. The resulting product contains an enormous quantity of minute sealed air cells, giving a good insulating efficiency combined with a moderate weight and

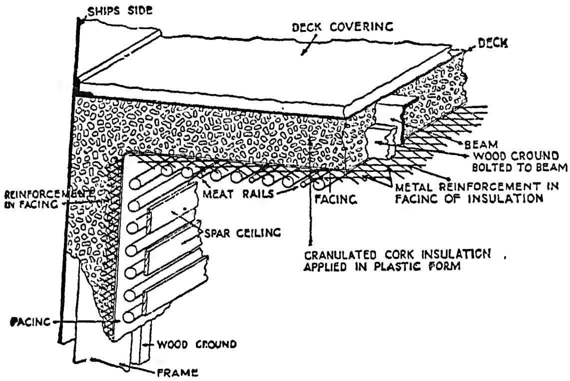

Fig. 17.—"Plastic" System of Insulation

high mechanical strength. The slabs and bricks are obtainable in both hollow and solid form. A cheap and efficient insulation can be secured by using these bricks in combination with other insulating materials.

The following figures give some idea of the heat leakage through various insulations. The figures give the heat leakage in B.Th.U. per 24 hours per square foot of surface for each degree F. difference of temperature between the two sides of the insulation.

$2\frac{1}{2}$-in. silicate cotton with one layer of $\frac{7}{8}$-in. boards and waterproof paper on each side	3·62
4-in. silicate cotton with one layer of $\frac{7}{8}$-in. boards and waterproof paper on each side..	3·0 to 3·5
4-in. silicate cotton and two layers of $\frac{7}{8}$-in. boards and waterproof paper on each side..	2·25
4-in. granulated cork with two layers of $\frac{7}{8}$-in. boards and waterproof paper on each side	2·0
8-in. granulated cork and two layers of $\frac{7}{8}$-in. boards and waterproof paper on each side	1·75
3-in. cork slab with one layer of $\frac{7}{8}$-in. boards and waterproof paper on each side	2·0 to 2·25
6-in. cork slab with one layer of $\frac{7}{8}$-in. boards and waterproof paper on each side	1·1
Brick wall with two layers of 3-in. hollow tiles, 4-in. silicate cotton between the two layers of tiles	0·7

CHAPTER III

Ice-making

One of the earliest uses to which refrigerating machines were put was the manufacture of ice. Three systems of ice-making were developed, namely:

1. The can system;
2. The cell system;
3. The plate system.

Of these the can system has by far the greatest application at present, while the cell plants are out of date.

The can system consists of a number of cans containing fresh water which are placed in a tank containing cold brine. The tank is often made of steel plates riveted together, with the refrigerator coils arranged down the sides and centre of the tank, but partitioned off from the remainder of it. Cross partitions are fitted at one end, having circular holes in them in which propellers revolve and keep the brine in circulation over the coils and cans. The cans are made of iron galvanized after completion, and taper on all sides towards the bottom to facilitate withdrawal of the ice when required. Particular care must be taken with the cans to see that the sides are not bulged, as if this occurs the ice block cannot be withdrawn without excessive wastage. The cans must also be maintained tight, otherwise brine will enter the ice and spoil it. The cans are, except in small plants, arranged in frames as shown in fig. 19. The tank is spanned by an overhead travelling crane. When the ice is made the crane picks up a frame of cans and lowers them for a few minutes into the thawing tank. Warm water is circulated through the thawing tank, loosening the ice from the sides of the cans.

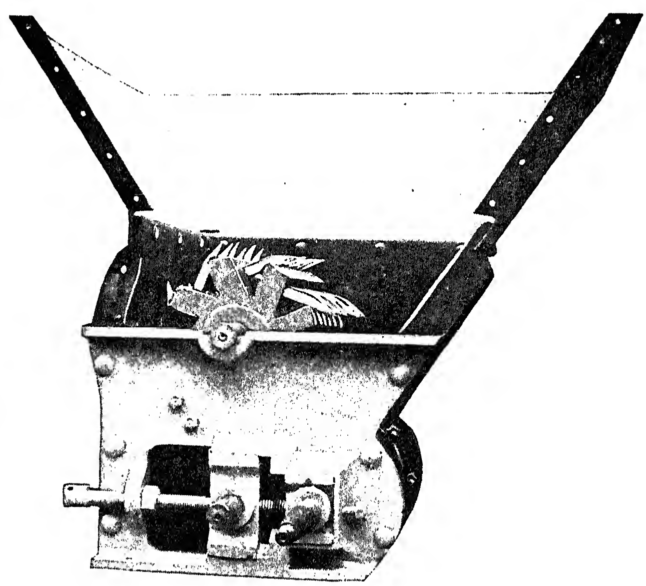

Fig. 18.—Ice Crusher

The crane now transfers the cans to a tipping cradle, which turns them over on their sides and slides the blocks out on to the floor. The empty cans are picked up by the crane and taken to the other end of the freezing tank to be refilled with fresh water. In large plants the cans are filled from a tank which has a number of nozzles with flexible pipes secured to a frame controlled by a lever. There is one nozzle for each can in a frame. By moving the lever the nozzles are lowered into the cans. The tank contains sufficient water to fill one frame of cans. All the foregoing features are well shown in fig. 19. In small plants the cans are filled individually by a hose and automatic valve. This is placed in the can and the water turned on. When the can is full a float is lifted and shuts off the water automatically.

The ice-making tank must be insulated on one or other of the systems already described. The tanks are sometimes made of wood and sometimes of concrete waterproofed with bitumen. In large plants the frame containing the cans may be fitted with wheels which run on rails attached to the tanks. At the end of the tank remote from the thawing tank, plungers are fitted. These plungers bear against the frames, and whenever a frame of cans is removed for thawing off, the plungers push the remaining frames towards the thawing-off end, leaving a blank space at the remote end for the new cans of fresh water. The cans thus travel through the freezing tank in definite rotation. This system is advantageous in large plants, but is too expensive for small ones.

The time required to freeze can ice may be obtained from the formula:

$$\text{Time in hours} = \frac{n(n+1)}{2}$$

where n is the thickness of the ice block.

Ice 6 in. thick would thus require 21 hours' freezing. The usual brine temperature is about 18° to 20° F., and about 120 to 150 ft. run of $1\frac{1}{4}$-in. pipe are used per ton ice-making capacity. The number of cans required depends upon the thickness of the ice and the time for freezing. The longer it takes to freeze, the more cans must be allowed per ton of ice required per day. Ordinarily fourteen 300-lb. cans may be used per ton of ice made per day. If the pipe surface or number of cans is reduced to save first cost, then a lower brine temperature must be carried to quicken up the freezing and give the desired output. This means greater operating expense.

When removed from the cans the ice blocks may be crushed for immediate use, sawn into smaller blocks, or placed in store. Fig. 18 shows a crusher. The upper edge of the hopper should be placed level with the floor, so that the ice blocks may slide into it. The outlet for the crushed ice is at the bottom, and from here it is conveyed by a shoot and placed in bags, barrels, or carts for delivery. If the crushed ice is for fishing vessels it may be taken direct to the quay-side and placed on board by band conveyors. Few mechanical contrivances are required for transporting the

ice blocks themselves, as they slide readily along inclined shoots. A lift may be necessary to give them the initial elevation requisite if the blocks have to go some distance.

The ice store is simply a large insulated chamber capable of holding the output of the plant for several weeks running. The store is usually cooled by direct-expansion pipes placed on the roof only. The stores are often of great height, some English ones being 35 ft. high, while in America heights of 40 and 50 ft. are used. The room is filled as completely as possible, leaving a space of about 12 in. between the sides of the store and

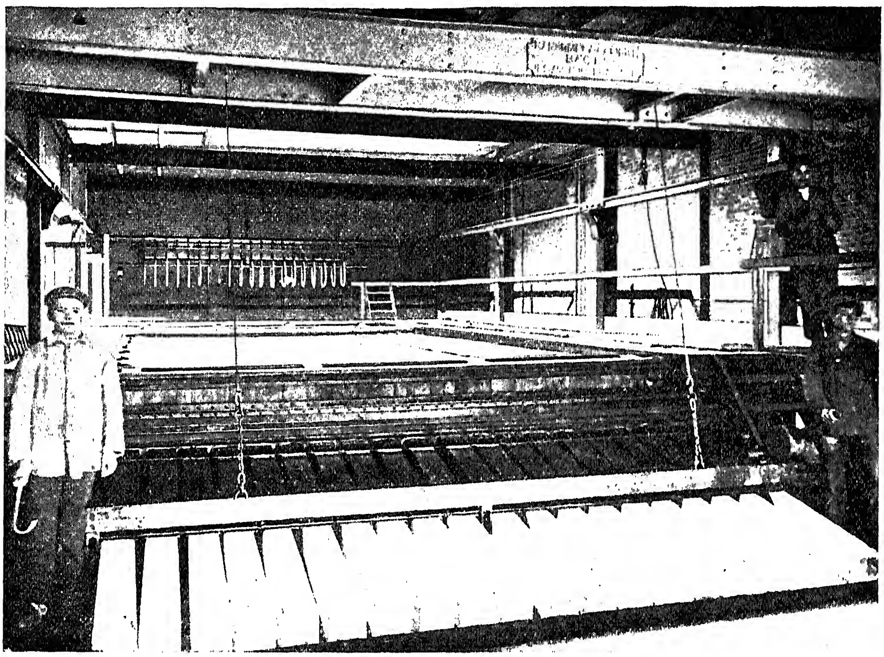

Fig. 19.—General Arrangement of Can Ice Tank

the ice to allow the cold air to circulate. Great care must be taken with the floor insulation and foundations to support the weight of ice in the store. The ice blocks, if dry before storing, may be piled directly on top of one another without any packing between them, and there need be no fear of their sticking when it is required to break them out. The blocks should therefore remain in a cold antechamber some time before going into the store itself. The insulation must be of a particularly waterproof nature as it is peculiarly liable to become wet. The author has not found anything so good for such situations as the plastic insulation previously mentioned.

Can ice is made in two kinds, clear (or crystal) and opaque. All water contains dissolved gases which are liberated at the moment of freezing. If the water being frozen is still the gas bubbles are imprisoned in the ice, giving it a white and opaque appearance. Such ice is only used for crush-

ing purposes, and being honeycombed is soft and rapidly melts. If the water is agitated during freezing the bubbles are washed away, producing a clear hard block of ice suitable for domestic and other purposes.

In small plants the ice store may be conveniently located beneath the ice tank. The bottom of the tank then projects into the store and keeps it cool.

With the plate system the ice tank is fitted at regular intervals with flat

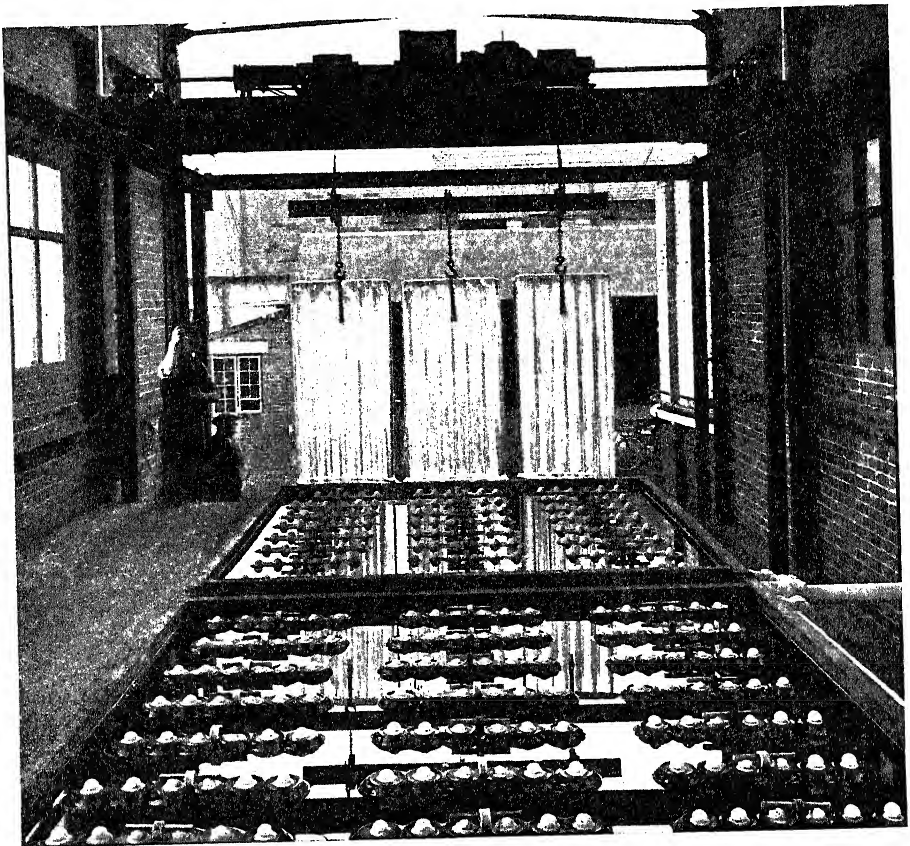

Fig. 20.—General Arrangement of "Pluperfect" Ice Tank

coils which are placed between metal plates. The ammonia passes through the coils, the water to be frozen being in contact with the plates. This system produces large blocks of ice of very good quality, but the time of freezing is long, requiring a large tank capacity for given output. This means high first cost, and there are comparatively few plants working on this system.

The latest and most promising development in ice-making systems is the Pluperfect. An ice tank on this system is shown in fig. 20. Such a tank is fitted with a number of single grids of tubes projecting upwards from the bottom of the tank. The number and spacing of these vary with

the quantity and size of block required. A usual size of block is 6 ft. by 3 ft. 9 in., weighing about 11 cwt. The grids themselves are made out of D-section tubes welded together along their flat faces, each pair being about 1 in. outside diameter. The top ends have a cap welded on them serving as a return bend. The lower ends are coupled to the next grid. The ammonia is passed through these pipes, causing the water to freeze around them until it forms a solid block. When the block has reached the desired size it is loosened from the vertical tubes by passing through them comparatively warm liquid ammonia on its way from the condenser to other tanks. The time on this system to produce a block of the size previously mentioned (about 10½ in. thick) is about 41 hours, while on the old plate system it would be about four to five days. Messrs. The Pluperfect Company state that a 50-ton plant requires about 2½ to 3 tons coal per day, or say 17 to 20 tons of ice per ton of coal. This has been exceeded, a plant in actual operation giving 28 tons ice per ton anthracite. An electrically operated plant has given 1 ton of ice for an expenditure of 56 units of electricity.

The author has had a small cell plant worked by CO_2 compressors which gave 17 to 19 tons ice per ton of anthracite used, average working.

CHAPTER IV

Cold Storage

In designing a cold store the general arrangement and disposition of the rooms is always governed by transport considerations, that is to say, the means of ingress and egress for the goods to be stored and the nature of the site. These goods may come by road, rail, or water, or more usually by a combination of two or all of these ways. Rapid handling is essential. Carts and railway wagons should come alongside a loading platform raised some 3 ft. above the ground. This platform should be provided with a sun screen. If large cargoes are to be landed from ships, use is made of various types of conveyors. The goods are taken into a receiving-room for sorting. This is usually on the ground floor, but in some stores it has been located on the top floor. In such cases the goods must all be carried to the top of the building and then lowered down inside to the various storage floors. The object of this design is to prevent the loss of refrigeration which ordinarily occurs with doors opening into each room from corridors. As, however, such a system increases the handling required, and seeing that even a large store does not require much power to keep it cool, the advantage of such a design is by no means demonstrated. In any case lifts or elevating conveyors must be provided, and these should be in duplicate. A stairway is also required giving access to the various floors.

All the cold rooms must be adequately insulated on one or other of the systems already described. Every room should be provided with an air lock or anteroom, so that there is never a direct communication between the cold room and the outside air when the doors are opened. Particular

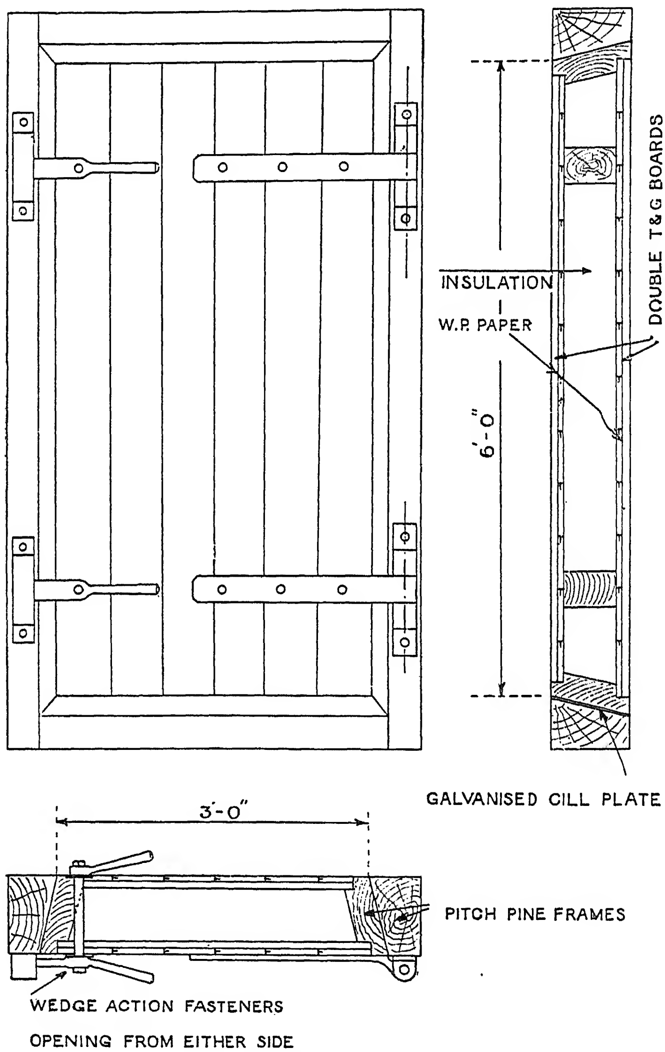

Fig. 21.—Insulated Door and Frame

care must be paid to the doors, of which a typical design is shown in fig. 21. The main timbers should be of pitch pine. The door is bevelled to fit the frame, and the edges are covered with felt to make a tight joint. The interior of the door is filled with insulation. The sill of the door is fitted with a galvanized plate to prevent damage to the woodwork when trucks are wheeled through. The door is carried on heavy strap hinges, and

very large doors are frequently supported by a runner wheel. The doors are secured by wedge-type fasteners and all ironwork is galvanized. All door fasteners should be capable of being operated from either side of the door, to prevent men being accidentally locked in the cold room. No windows are of course permissible in the cold rooms, and these must be electrically lighted. The light fittings must be of the waterproof type.

With regard to the cooling of the rooms there are three ways in which this may be accomplished:

1. By direct-expansion piping;
2. By brine piping;
3. By cold-air circulation.

In the first the rooms are fitted with pipes on the sides and roof through which the ammonia is passed directly. This method requires the least amount of piping, and is therefore cheap. If, however, the machinery is stopped for any length of time, the rooms rise in temperature, as there is no reserve of cold in the pipes to absorb the heat leaking in.

In the second method the rooms are arranged with pipes as before, but cold brine is circulated through the pipes instead of ammonia. The pipes are always full of cold brine, and the rooms keep cool even should the machinery be stopped. This method gives the steadiest temperature, but requires most piping, as a refrigerator to cool the brine is needed in addition to the piping in the rooms themselves. Brine-circulation pumps are also required. These may be of any type, the centrifugal having many points of advantage. It is strongly to be recommended, however, that the brine pumps should be driven independently of any other machine, so that they can be worked at any time.

In the third method an air-cooling refrigerator is arranged outside the cold rooms. Each room is fitted with air ducts leading to and from the air cooler. The air circulates through the rooms and back to the cooler to be recooled. There is no piping in the rooms, and therefore no reserve of cold to guard against a rise of temperature when the machine is stopped.

In the first two methods the cold is distributed throughout the rooms by the natural circulation of the air set up by the cold pipes, and the coils should be arranged to give full play to this effect. There is, however, no ventilation or change of air in the room except such as may occur by opening of doors. Some classes of goods give off gases freely, and if the air of the room becomes foul, odours will be transmitted from one kind of produce to another. This may cause heavy loss through goods being spoiled. If the cooling is done by air circulation good ventilation is secured, but where air from all the rooms is circulated over a common air cooler there is a risk of tainting differing kinds of goods. Usually a combination of air cooling and piping will be found best.

Proper ventilation is an essential but often overlooked point. The ideal system would be to ventilate the rooms with fresh air drawn from

outside, this of course being cooled down to the room temperature before being blown into the room.

A second and also frequently neglected point in cold-storage work is the humidity or moisture content of the air. If this is too dry it will take up moisture from the goods and cause them to shrink excessively. On the other hand, if it is too wet it will deposit moisture on the goods, favouring the growth of moulds. The humidity of the air should be regularly tested and kept at the right point for the class of goods being stored.

When small rooms are cooled by direct-expansion pipes, these are frequently led through larger pipes or drums filled with brine. This becomes cooled when the machine is at work, and keeps the room cold when the plant is stopped. Instead of drums, flat vertical tanks called brine walls are sometimes used.

The height of ordinary cold-storage rooms is usually about 7 or 8 ft., whilst rooms for hanging meat may go up to 10 ft. Such rooms will require to be fitted with overhead meat rails to take the meat hooks. In some cases, especially in cold rooms connected with abattoirs, an elaborate system of runner rails is fitted. The meat hooks are fitted with wheels which run on the rails, thus facilitating handling.

The brine used in refrigerating work is made from calcium chloride. Common salt should not be used, as it cannot be cooled to as low a temperature as calcium brine without freezing. A useful figure to remember is that 1 gall. of brine absorbs $8\frac{1}{2}$ to 9 B.Th.U. in changing its temperature 1° F. The published data regarding the freezing-points of brines of different strengths is so variable that it is not reproduced here. Calcium chloride brine is usually used at a density of 44 Tw.*

TABLE VIII

SQUARE FEET OF DIRECT-EXPANSION PIPING REQUIRED PER 100 CUBIC FEET OF STORAGE SPACE

Size of Room, Cubic Feet.	Temperature Fahrenheit.			
	0°	10°	20°	30°
1,000	31	8	5	$3\frac{3}{4}$
10,000	$18\frac{1}{2}$	5	3	$2\frac{1}{4}$
50,000	$13\frac{1}{2}$	$3\frac{1}{2}$	$2\frac{1}{4}$	$1\frac{3}{4}$
100,000	$11\frac{1}{2}$	$3\frac{1}{4}$	2	$1\frac{1}{2}$

The above surfaces are mean surfaces, i.e. average of the internal and external surfaces.

For rooms less than 1000 c. ft. the best practice is to put in as much piping as possible. The above figures are based on good insulation.

With brine circulation these figures will require to be increased by

* 44 Tw. (Twaddle) ≡ a density of $1 + \frac{0.44}{2}$, i.e. 1·22.

50 to 100 per cent, depending upon the conditions. This is because the temperature difference between the room and the cooling medium is usually less with brine circulation than with direct expansion.

TABLE IX

AMOUNT OF COOLING TO BE DONE, IN B.TH.U., PER 24 HOURS PER CUBIC FOOT OF STORAGE SPACE

Size of Room, Cubic Feet.	Temperature Fahrenheit.			
	0°	10°	20°	30°
1,000	550	190	100	75
10,000	300	130	75	55
50,000	130	75	50	35
100,000	100	50	30	20

The above figures assume good insulation and average atmospheric conditions in this country.

One ton of refrigeration is approximately equal to 322,000 B.Th.U. per 24 hours, and is also approximately equal to the freezing of 1 ton of miscellaneous goods.

STORAGE TEMPERATURES FOR VARIOUS PRODUCTS

Apples	30° to 35° F.
Bananas	35° ,, 40°
Beef (chilled)	29° ,, 30°
Beef (frozen)	18°
Beer	35° ,, 36°
Butter	14°
Cheese	30°
Eggs	30°
Fish (after freezing)	15° ,, 18°
Fish (fresh)	28°
Game	15° ,, 20°
Milk	32° ,, 36°
Potatoes	34° ,, 38°
Poultry (frozen)	10°
Poultry (fresh)	30°

CHAPTER V

Marine Refrigeration

In marine work refrigeration is applied to three general purposes:

1. Preserving ships' provisions;
2. Carriage of fruit cargoes;
3. Carriage of meat cargoes.

Plants of the first class are of small size. Fig. 22 shows a typical layout of rooms for a cargo steamer. The cooling is done by brine piping or direct expansion. The rooms are insulated as already described and fitted with insulated doors. Loose filling and boards is the usual insulation for ship work, but even here a faced insulation is coming into favour. This is the more desirable, as shipboard insulation has more chance of harbouring vermin and becoming wet. The provision chambers are fitted with rails for the hanging of meat and shelves for other goods. These latter are now frequently made of galvanized iron.

With regard to the carriage of fruit cargoes, a cold-air circulation system is usually adopted in such cases. Air circulation is desirable on account of the gases which fruit gives off. The temperature required is not very low, being about 40° F. The insulation, therefore, is not very thick. Air trunks are constructed along each side of each hold by erecting a partition of boards. This partition has a number of openings or ports in it fitted with sliding shutters in teak frames. The cold air from the refrigerator is blown down the air trunks on one side of the hold, passes through the openings or ports and across the hold, cooling the fruit as it does so. It then passes through the ports into the air trunk on the other side of the hold, and is drawn back to the refrigerator for recooling. To facilitate loading of the fruit, especially bananas, the hold is divided up into a number of bins. This is done by setting up posts at intervals of, say, 10 ft. These posts support spars of, say, 5 in. by 1 in. timber, openly spaced, thus cutting up the hold into rectangular bins in which the fruit is loaded. The loading must be done so that there is a free circulation of air through the cargo.

Refrigerating plants for the carriage of meat cargoes are of much more elaborate construction. Two kinds of cargoes are carried, namely, frozen and chilled. Chilled meat is a delicate product requiring careful handling and has a life of about five to six weeks. Frozen meat is carried at a temperature of about 18° F., and a slight variation of temperature does not necessarily do any harm. On the other hand, chilled meat is carried at 29° to 30° F., and every endeavour is made to keep the temperature steady within a quarter of a degree. It is this latter fact that accounts for the elaborateness of such refrigerating installations. Wood grounds are secured to the frames and beams and also laid on the tank top. A double lining of boards is nailed to the grounds, with waterproof paper between them, and

the space between the boards and ship's structure filled with silicate cotton or granulated cork. Over the bilges the insulation is movable, being made in the form of plugs with bevelled pitch-pine framework similar to a hatch plug. The thickness of the insulation is varied to suit the needs of the

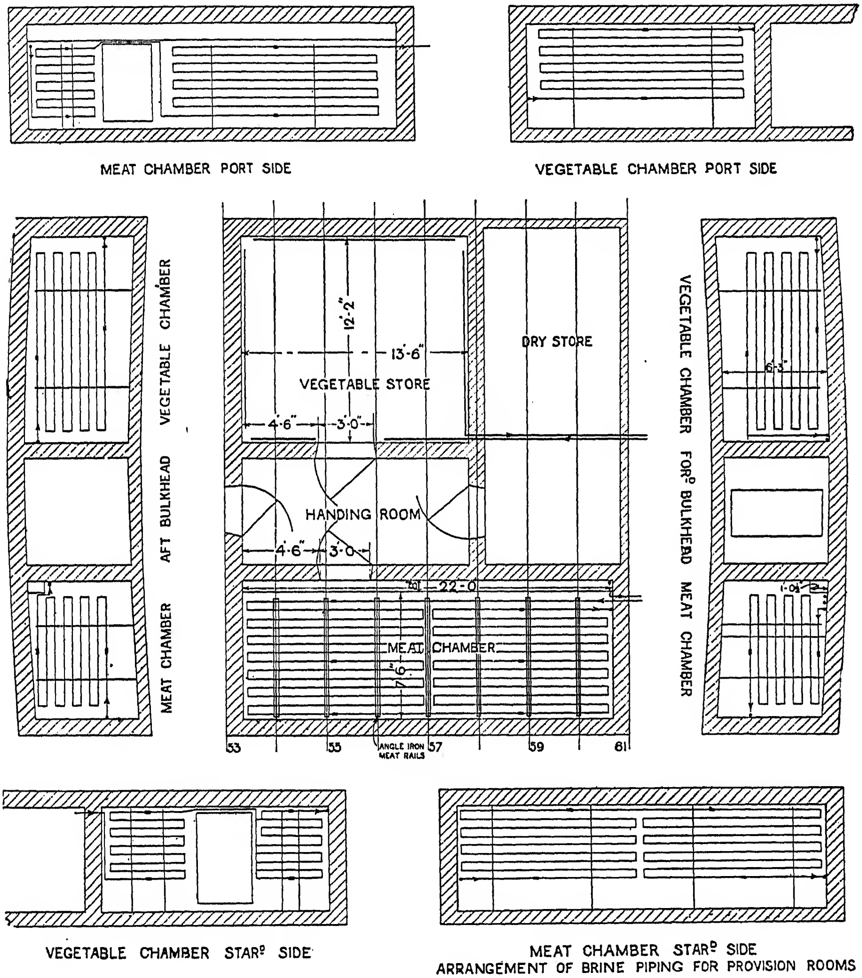

Fig. 22.—General Arrangement of Ship's Provision Chambers

Capacity of meat chamber = 1118 c. ft. Ratio of brine piping 1·4 to 1.
Capacity of vegetable chamber = 1004 c. ft. Ratio of brine piping 2·33 to 1.

situation, being thicker on hot surfaces such as the engine- and boiler-room bulkheads than elsewhere. The thickness will vary from 10 to 14 in. The hatches are fitted with insulated plugs resting on heavy bevelled pitch-pine bearers secured to the hatch coamings. The hatchways are also trunked round or enclosed by an insulated partition which has large doors to give access to the holds. In order to utilize all space as fully as possible, the

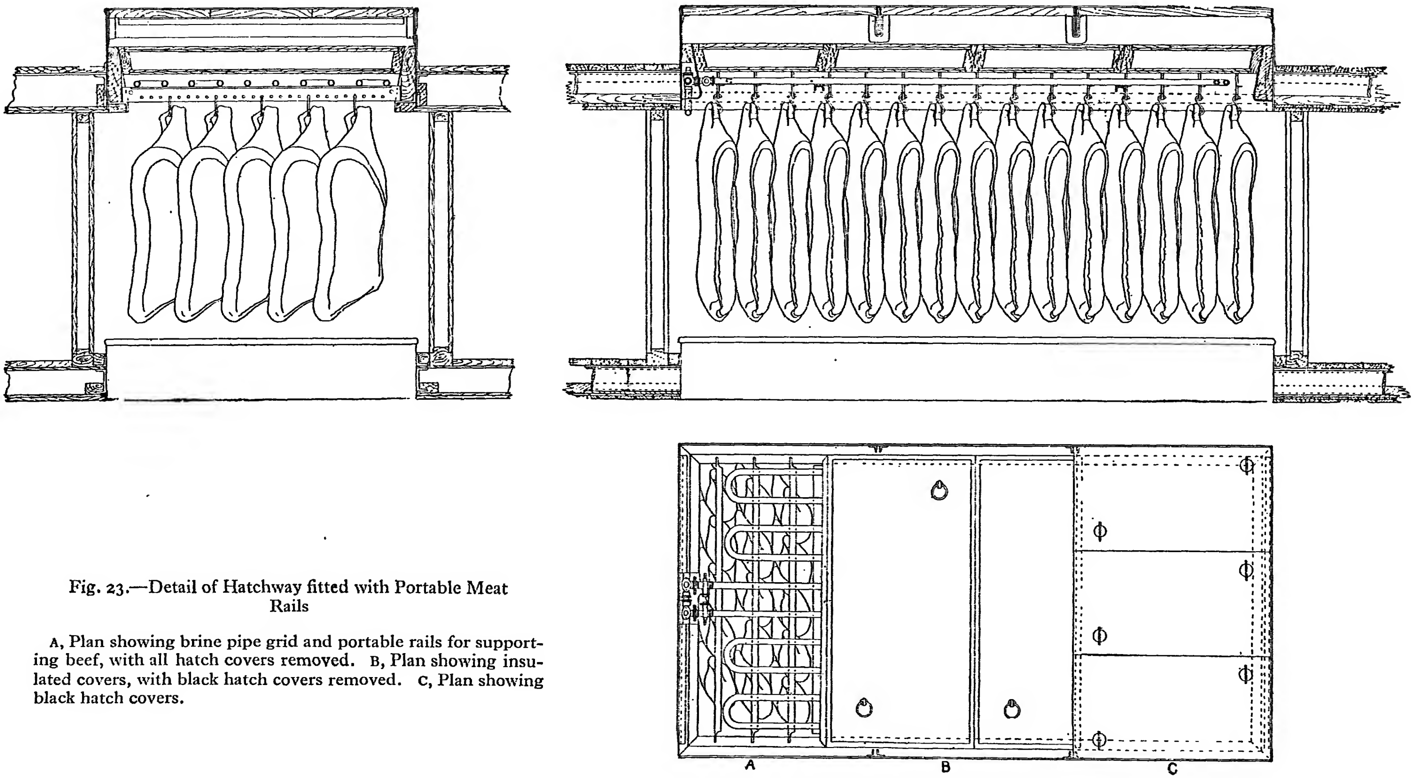

Fig. 23.—Detail of Hatchway fitted with Portable Meat Rails

A, Plan showing brine pipe grid and portable rails for supporting beef, with all hatch covers removed. B, Plan showing insulated covers, with black hatch covers removed. C, Plan showing black hatch covers.

hatchway spaces are fitted with portable cooling coils and meat rails, and as the ship is loaded these spaces are also filled up. Fig. 23 shows a typical hatchway illustrating the points mentioned.

Frozen carcasses may be stacked upon one another, but chilled meat must always be hung. We therefore find that frozen meat is carried in the deep lower holds and chilled meat in the 'tween decks. The meat rails are of galvanized iron tube. The cooling of the holds is always done by circulating cold brine through the pipes. Two different temperatures of brine are required, one for the chilled holds and the other for the frozen. These are sometimes produced by using separate refrigerators, one or more compressors working on each refrigerator according to the amount of work to be done at the differing temperatures. Separate brine pumps are used which pump the brine through large mains to a distributing room, and from this room any hold can be supplied with either freezing or chilling brine at will. The grids in the various rooms are coupled together in series to a length of about 1300 ft., and then a return pipe is run back to the distributing room. Each brine lead and return is provided with shut-off valves and a thermometer, enabling the supply to each circuit to be regulated properly. The brine returns may discharge into an open tank, thus being visible, or into a closed tank. In some cases flow meters are fitted to ascertain the quantity of brine flowing in each circuit, but these are not universal. The pipes in the holds become coated with snow, and this has to be removed at the end of the voyage after unloading. This is done by circulating warm brine through the pipes by a special pump. Sawdust is scattered over the floor to absorb the melting snow and prevent it wetting the insulation.

An alternative method of producing the two brine temperatures required is to use an attemperator. With this system the refrigerators cool all the brine to the lowest temperature required, namely, that for the frozen meat. The chilling brine is made by mixing a small quantity of frozen brine with the comparatively warm brine returning from the holds, the amounts being proportioned to give a mixture of brine at the correct chilling temperature. There is no difficulty in doing this, and fig. 25 shows an attemperator or mixing valve designed for this purpose. The freezing brine enters at A and the warmer return brine at B. The mixture reduced to chilling temperature leaves at C. The valve D varies the proportions of the mixture until the thermometer F indicates the correct chilling temperature.

In all installations where brine pipes pass through engine, boiler, and bunker spaces they must be very carefully insulated, and the outside of the insulation sheathed with galvanized plate to protect it from damage.

Some experiments have recently been made in carrying fruit from Australia in ships fitted out for meat cargoes. The cooling is in this case done by circulating cold brine through the pipes, and not by circulating cold air. All air trunks are thus dispensed with, and the space which they would have occupied becomes available for cargo. The success of such experiments also enables ships fitted for meat carrying to be put into the fruit trade, when circumstances render it desirable, without altera-

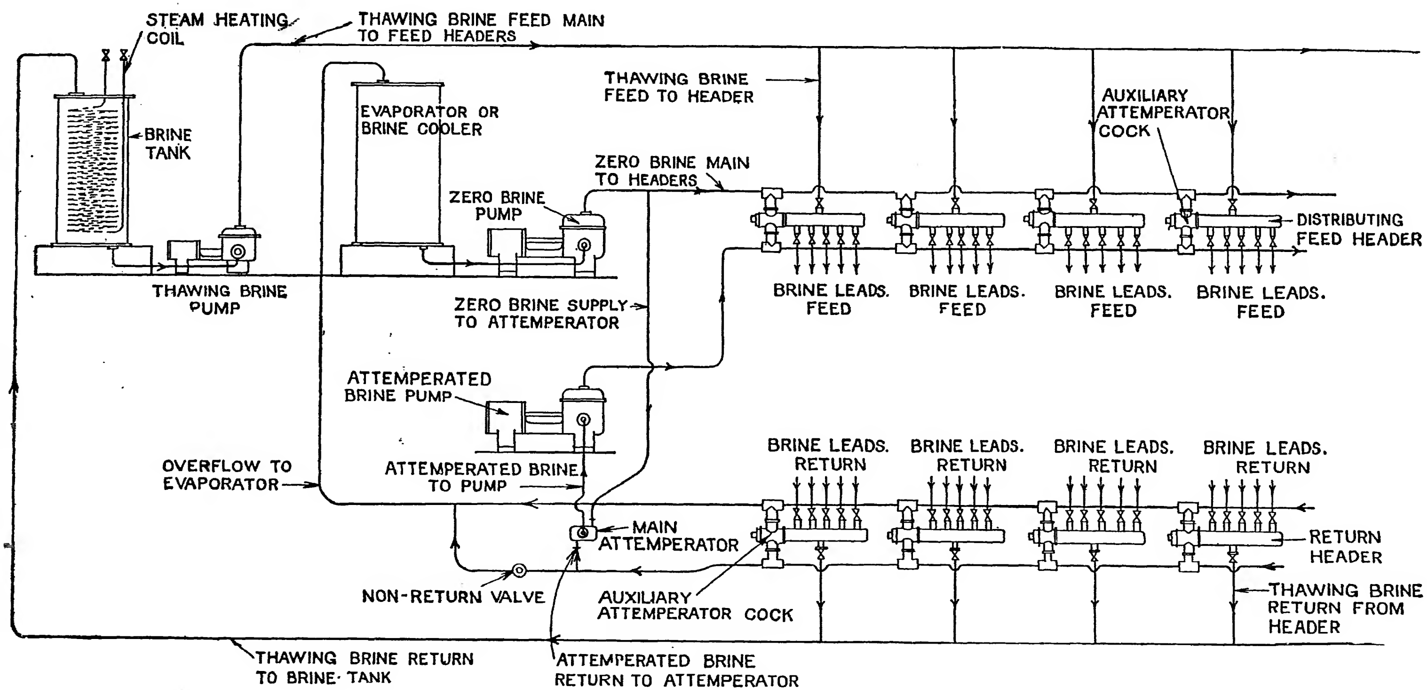

Fig. 24.—Diagram Arrangement of Webb's Patent Attemperated Brine System

tion. To prevent condensation from the pipes dripping on to the fruit, the cold brine must be at a temperature low enough to freeze the condensation on to the pipes, and care must then be taken not to place the fruit so close to the pipes as to result in it becoming frost-bitten. There being no air circulation, ventilation has to be carefully given in the ordinary way when required.

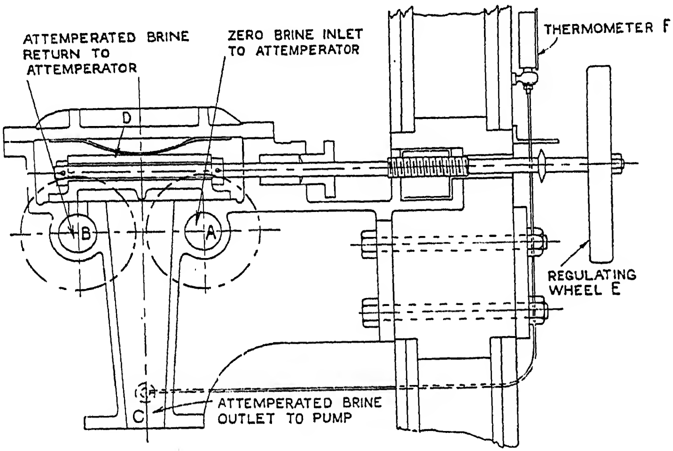

Fig. 25.—Main Brine Attemperator

Lack of space has prevented dealing with the more special uses of refrigeration in the chemical industry, &c.

The author is indebted to the following firms for kindly supplying illustrations: Messrs. Ransomes & Rapier for figs. 2, 13, and 19; L. Sterne for figs. 4 and 15; The Liverpool Refrigeration Company for figs. 5, 9, 22, 23, and 24; Haslam for fig. 10; J. & E. Hall for figs. 11 and 12; The Noel Insulation Company for fig. 17; Pertwee & Back for fig. 18; The Pluperfect Refrigeration Company for fig. 20; Sulzer Brothers for fig. 8; Brotherhood for figs. 6 and 7.

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